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EXPERIMENTAL RESEARCHES

ON

THE FLOW OF STEAM THROUGH NOZZLES AND ORIFICES

TO WHICH IS ADDED A NOTE ON

THE FLOW OF HOT WATER

BY
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(EXTRAIT DES ANNALES DES MINES, JANUARY 1902)

AUTHORIZED TRANSLATION
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PREFACE.

In the steam-turbine we are confronted with an entirely new series of phenomena. The knowledge of steam accumulated by familiarity with the reciprocating engine is of little use in designing this newest yet oldest of prime movers. Nor can the cut-and-try methods that have brought the reciprocating engine to its present and, it may almost be said, its final stage of excellence be employed in the steam-turbine. Modern commercial conditions are such that design must be based on ascertained physical laws.

The development of the steam-turbine within the last few years has been remarkable. It has reached a position not only of equality with the best modern engines, but its high efficiency, its lower first cost, smaller size, and ease of operation have placed it in the very forefront of steam-engineering practice.

This rapid advance has necessitated the re-investigation of steam from the point of view of velocity instead of pressure, of kinetic energy instead of potential, of the friction of rapidly moving bodies against a vapor instead of a lubricated surface, of the balancing of parts so that they shall revolve about their centre of gravity, and of numerous other questions which are but beginning to make themselves felt. The available data are few, and practically all the information is scattered widely through the pages of French or German scientific periodicals.

Of all these, the fundamental problem is the proportioning of the nozzles or passages which shall permit the steam to transform its pressure into vis viva. While there exists considerable information on this subject throughout the scientific press, very little is reliable or in such shape as to be of practical value. The experiments undertaken by Prof. Rateau on this most important feature of steam-turbine design were made when he was beginning his study of the steam-turbine and evidence a care and precision of scientific research coupled with a keen appreciation of the practical that give a report of his results a value to the designer too often wanting in the ordinary laboratory investigation.

In the hope that these results may be of some service to those who have neither the time nor the inclination to study them in the original, this translation has been made.

H. B. B.

CHICAGO, September 1904.

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EXPERIMENTAL RESEARCHES

ON

THE FLOW OF STEAM THROUGH CONVERGENT NOZZLES AND ORIFICES IN THIN PLATES.

1. The operation of steam turbines depends upon the flow of steam at a high velocity. In order to be able to calculate these machines rationally it is necessary to have a complete knowledge of the phenomenon of the flow of steam. As only brief, incomplete and sometimes erroneous ideas are found in the works on mechanics and thermodynamics, I undertook extended and precise experimental researches in order to verify the thermodynamic principles involved.

The results of these researches, made in 1895–96, are here published. We shall see that the theoretical formulæ, correctly interpreted, are exactly confirmed, and also that the experimental results allow of determining very closely the mechanical equivalent of the heat unit.

Before describing the method employed I will outline the theory of the phenomenon.

2. **Theory.**—The flow of elastic fluids through nozzles differs considerably from the flow of liquids as soon as the ratio of the pressure in the exhaust to the initial pressure $\frac{p}{P}$ becomes appreciably less than unity.

If at any point in a nozzle

S be the area of the nozzle,
V the velocity of the fluid,
Q the volume of discharge,
W the weight discharged,
D the mean density of the fluid in the state in which it exists, i.e., homogeneous or heterogeneous, while passing the section S, then the volume of the discharge Q is at once given by $Q=VS$, and the weight of the discharge is equal to DQ and is related to the section S and the velocity V by the following formula:

$$W=DVS. \qquad \text{. . . . . . .} \qquad (1)$$

If D is constant, as in the case of liquids, the section S is inversely proportional to the velocity V, and consequently V always increases as S decreases, but this is no longer true in the case of elastic fluids. As the pressure falls the density decreases proportionally, while the velocity increases in such a manner that the product DV first increases, reaches a maximum, and then decreases. In the case of gases the maximum occurs for a ratio of the pressures equal to 0.52, and for steam when the ratio is in the neighborhood of 0.58 whatever the initial pressure may be.[1] The calculation of this maximum will be given later. It follows that when the exhaust pressure is lower than $0.58P$ the discharge nozzle, at first convergent, should then diverge, if it is desired that the steam shall continue to expand so as to reach the speed corresponding to the fall of pressure from P to p; and the ratio of the final section of the nozzle S_1 to the section at the throat S should vary with the ratio of the pressures. At the throat the pressure is always equal to $0.58P$, and the velocity, which depends on P only, is the same as sound would have in the fluid in the state in which it exists at that point of the nozzle, as has been shown by Hugonoit (Comptes Rendus de l'Académie des Sciences, vol. 103).

With a given nozzle, if the pressure p into which the steam is discharged is lower than the value p_1 corresponding to the ratio $\frac{S_1}{S}$ of the sections of the mouth and the throat of the nozzle, the pressure of the steam at the mouth of the nozzle will not be p, but p_1, which is in a fixed ratio to the initial pressure P. At the moment of leaving the nozzle, the steam entering suddenly into a space where the pressure is lower than in the mouth of the nozzle immediately expands and the jet takes a paraboloid form. The enlargement of the jet ceases when the back pressure reaches the value p_1.

---

[1] The value of this ratio which makes DV a maximum appears, however, to depend a little on the value of P in the case of steam.

The discharge is independent of the back pressure when that is less than $0.58P$. On the contrary, when p is larger than $0.58P$ the discharge depends both on P and p. It is necessary to distinguish two very different cases, therefore· in the first the calculation of the discharge depends only on P and the formula is simple; in the second it depends both on P and on p.

For both cases, however, it is the most contracted section of the nozzle (the throat if it is convergent-divergent, or the mouth if it is simply convergent), which enters into the calculation of the discharge.

The above is only applicable, strictly speaking, to nozzles properly so called. In the case of orifices in thin plates the coefficient of contraction K, which varies with the ratio $\frac{p}{P}$ of the pressures, complicates the phenomenon. We shall see later on the experimental values that were obtained for this coefficient.

3. **Equation of Velocity.**—The velocity V of the steam can be calculated by two different methods, either by the ordinary mechanical method having given the relations between the specific volume s and the pressure p, or by thermodynamics having given the thermal constants of the steam, which can be obtained from Regnault's steam tables.

The first method results in the general formula usually credited to Weisbach, but already pointed out by Wantzel and Saint Venant in 1839,

$$\frac{V^2}{2g}=\int_p^P v.\,dp, \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

which can be integrated when the relation between v and p is known.

It is always assumed that in the flow of steam through nozzles the expansion is adiabatic because the steam remains so short a time in the nozzle, some 0.0001 of a second, that it cannot give up or receive an appreciable quantity of heat. Now in the case of adiabatic expansion of initially saturated steam Zeuner has shown[1] that within large limits of pressure we have approximately

$$pv^\gamma=\text{constant}, \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

as in the case of gases, but with $\gamma=1.035$ instead of 1.41. Putting the value of v calculated by this equation in the preceding equation, V is obtained as a function of P and p. Then from Zeuner's empirical formula $D=0.587P^{0.94}$, which gives the density of steam as a function of its pressure, we have the following formula for the discharge when $P=0.58P$:

$$W=15.26P^{0.97}, \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

---

[1] Mechanical Theory of Heat.

W being the weight of steam discharged in grammes per second per square centimetre of orifice.[1]

This formula was given by Dr. Grashof (Theoretische Maschinenlehre, Vol. I, III).

By the second method, developed by Zeuner, the kinetic energy $\frac{V^2}{2g}$ of unit weight of the fluid should be equal to the energy represented by the entropy diagram when the expansion is adiabatic from P to p.

Putting T_0 the temperature of saturated steam corresponding to P, T_1 that corresponding to p, in the entropy diagram (Fig. 1), with entropy ϕ as abscissæ and absolute temperature T as ordinates, AE and DF are the isothermals at the temperatures T_0 and T_1. AD is the curve of entropy for water, and EF that for saturated steam. Let B be the point corresponding to the state of the steam as it enters the nozzle. Now the relative weights of liquid and vapor in the mixture represented by the point B are proportional to the lengths BE and BA. (We are here supposing for the sake of greater generality that the fluid is not initially in the state of saturated steam.) BC is the line representing the expansion of steam in the nozzle. If the expansion is adiabatic, BC is a

---

[1] In English units this formula becomes

$$W = 0.0165P^{0.97},$$

where W = pounds discharged per square inch of orifice per second and P = absolute pressure in pounds per square inch. —TRANS.

straight line parallel to the axis of temperatures. The total mechanical energy developed by the fluid during its complete expansion from the pressure P to the pressure p is represented on the diagram by the area of the trapezium $ABCD$. This area is given by

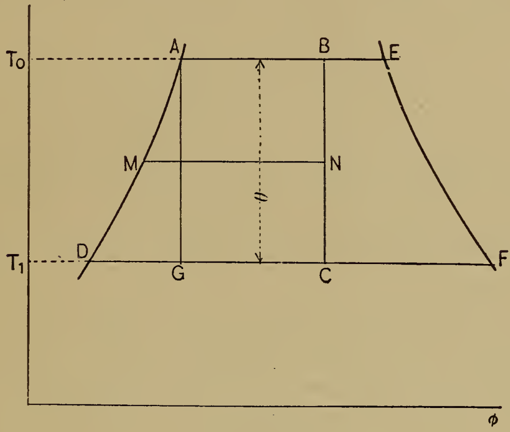

Fig. 1.

the simple expression θL, in which θ is its height (that is to say, the fall of temperature from T_0 to T_1) and L its mean length MN.

When the fall of temperature is small this mean length L can be taken in practice equal to the mean of the two bases AB and CD; but if the exact formula is required, the trapezium must be divided into the rectangle $ABCG$, equal to θAB, and the triangle with curvilinear hypotenuse AGD. If the steam is initially saturated, AB is equal to $\frac{r}{T_0}$, r being the heat of vaporization of the water at the temperature T_0.

Introducing E, the mechanical equivalent of heat, the formula for the velocity becomes

$$V^2=2gE\theta L, \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

and if the steam is initially saturated, we may write

$$V^2=2gE\left(\frac{r}{T_0}+\frac{C\theta}{2T}\right), \quad . \quad . \quad . \quad . \quad (6)$$

C being the specific heat of the liquid and T being nearly the arithmetic mean $\frac{T_0+T_1}{2}$ of the extreme temperatures.

This is the simplest form of the equation by which the velocity of the flow of steam can be calculated. It supposes a knowledge first of the temperatures corresponding to the pressures; second, the heat of vaporization of the liquid; third, the specific heat of the liquid. These values are given for water by Regnault's tables with an approximation more than sufficient in practice.

4. **Calculation of the Weight of Discharge.**—In order to calculate the weight of discharge, the velocity of flow, the density of the steam at the section of the nozzle under consideration, and the quantity of liquid formed during the adiabatic expansion must be ascertained. This liquid is entrained in the steam and probably makes with it a homogeneous mixture.

The density can be obtained from Regnault's tables by means of Clapeyron's formula. It has already been calculated and is generally found in a supplementary column added to Regnault's tables.

The quantity of liquid formed can be calculated from the entropy diagram, or can be taken from a chart, as I have shown in my Rapport sur les Turbines à Vapeur.[1] The proportion of water thus obtained is $1-x$, where x is the quality of the steam.

Neglecting the specific volume of the liquid compared with that of the steam (which is many hundred times larger), we have for the weight of discharge

$$W=\frac{DSV}{x}.$$

Conversely, being given a weight W of steam discharged by a nozzle, we can calculate by this formula the area necessary when the pressure has any given value p. An example of this calculation will be given later.

In the first place, however, it should be noted that the two methods employed in calculating the velocity and weight of discharge while differing in form are really identical, because it was from Regnault's tables that Zeuner deduced his empirical formula connecting the pressure and specific volume of a mixture of steam and water during an adiabatic expansion, so that both methods are based finally upon Regnault's tables; that is to say, on the experimental data of thermodynamics.

---

[1] Memoir presented to the Congrès International de Mécanique, 1900. Rev. de Mécanique, August 1900.

5. Profile of Nozzles for Steam. — In the following table is given the calculation of the velocity of flow of initially saturated steam from an initial pressure $P = 10$ kilogrammes per square centimetre.

The first column of this table gives the pressure p at which the speed is to be calculated.

The second column gives the fall of temperature θ; that is to say, the difference of the temperatures corresponding to the pressures P and p.

The third column gives the value of L; that is to say,

$$\frac{r}{T} - \frac{\theta}{2T'}, \quad \text{where} \quad T' = T - \frac{\theta}{2}.$$

The fourth column gives the velocity of flow V in metres per second calculated from the preceding formula (6).

In the fifth column the density corresponding to the pressure p is given.

The sixth column contains the product DV.

In the seventh column is given the proportion of steam remaining when the mixture has expanded adiabatically to the pressure p. The fraction x is calculated by means of the entropy diagram or by Zeuner's formula

$$\frac{r_1 x_1}{T_1} = \frac{r_0 x_0}{T_0} - C \log \frac{T_1}{T_0}.$$

In the eighth column the quotient x by DV is written, or rather, to avoid a number of zeros, 1000 times this quotient,

$$1000 \frac{x}{DV}.$$

Calculation of the Velocities and Theoretical Areas for the Flow of Saturated Steam.

Initial pressure 10 *kg. per sq. cm.*

R is the ratio of the radius of the area of flow corresponding to the pressure indicated in the first column, to the radius of the area at the throat of the nozzle.

p kg/cm² 1	θ 2	$\frac{r}{T}+\frac{\theta}{2T-\theta}$ 3	V m/sec 4	D 5	DV 6	x 7	$1.000\frac{x}{DV}$ 8	R 9
10	°C.							
9	4.521	1.0673	200.554	4.6160	925.757	0.9933	1.0730	1.2358
8	9.427	1.0728	290.347	4.1331	1200.032	0.9858	0.8215	1.0813
7	14.862	1.0791	365.629	3.6467	1333.338	0.9767	0.7325	1.0210
6	20.943	1.0860	435.417	3.1558	1374.012	0.9673	0.7041	1.0009
5.9	21.596	1.0868	442.330	3.1065	1374.100	0.9661	0.7031	1.00024
5.8	22.258	1.0875	449.188	3.0570	1373.169	0.9650	0.70276	1
5.75	22.594	1.0879	452.650	3.0322	1372.523	0.9646	0.70279	1.00007
5.7	22.929	1.0883	456.076	3.0075	1371.650	0.9642	0.70295	1.00015
5.6	23.610	1.0892	462.991	2.9580	1369.528	0.9632	0.70331	1.00042
5.4	24.999	1.0908	476.765	2.8589	1363.024	0.9607	0.70483	1.0015
5	27.899	1.0942	504.445	2.6596	1341.620	0.9562	0.71272	1.0071
4	36.078	1.1038	576.152	2.1572	1242.875	0.9432	0.75885	1.0392
3	46.088	1.1159	654.753	1.6466	1078.116	0.9279	0.86067	1.1066
2	59.326	1.1328	748.580	1.1250	842.150	0.9068	1.0768	1.2378
1	79.808	1.1591	878.121	0.5860	514.579	0.8755	1.7016	1.5563
0.8	85.884	1.1673	914.152	0.4749	434.131	0.8654	1.9934	1.6842
0.6	93.404	1.1775	957.490	0.3621	346.707	0.8535	2.4617	1.8718
0.4	103.425	1.1914	1013.470	0.2470	250.328	0.8374	3.3452	2.1820
0.3	110.148	1.2011	1050.140	0.1882	197.637	0.8259	4.1789	2.4386

The sections of a nozzle required to discharge a predetermined weight of steam will be proportional to this quantity. To obtain the absolute values, the actual discharge of steam must be known.

Examining the numbers in this column it will be noted that the areas decrease until p is equal to $0.58P$, and then increase indefinitely. In order to determine as exactly as possible the position of this minimum, we have made the intervals of pressure p in the neighborhood of the value $0.58P$ much smaller. By tracing the curve of the quantity $1000\,\frac{x}{DV}$ as a function of the pressure p, the position and the value of this minimum are determined with an approximation as close as is permitted by Regnault's tables.

I have made this calculation for the discharge from a number of initial pressures, and can state that the position of the minimum varies slightly with the pressure P around the value $0.58P$.

Taking the square root of the quantity $1000\,\frac{x}{DV}$ and forming the ratio of this square root to its smallest value, we obtain a quantity which we may call R, which will be proportional to the diameters or radii of a nozzle of circular cross-section of proper shape to discharge the steam. The values of R are written in the last column of the table. Plotting as abscissæ either p or some function of p, for example log p, and as ordinates the value of R, we can represent the longitudinal profile of nozzles for the flow of steam. Fig. 2 gives this curve by which the nozzles of steam turbines can be calculated rapidly for each particular case.

This curve is utilized in the following manner. Having the initial pressure P and the final pressure p to which the steam is required to expand, we take from the curve the ordinate corresponding to the abscissa indicated by the value of the ratio $\frac{p}{P}$. This ordinate is then the radius of the final section of the nozzle, the section at the throat having the minimum ordinate of the curve as its radius.

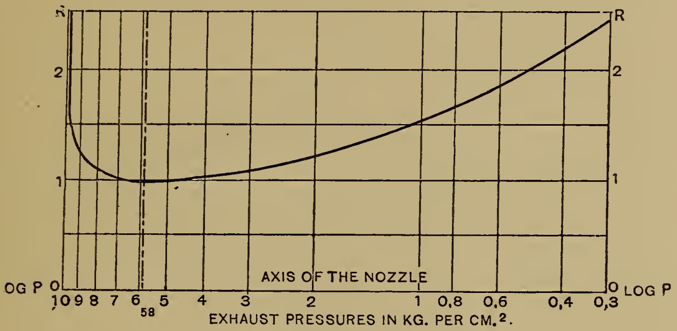

FIG. 2.

It is true the curve will be slightly different for different initial pressures, but for the needs of practice the variation can be neglected.

6. Practical Formula for Discharge.—When the nozzle is convergent and the final pressure p is less than $0.58P$ the discharge depends only on P, because then the pressure p at the mouth of the nozzle is always equal to $0.58P$. Calculating the discharge W per unit area of the mouth of the nozzle, or calculating the ratio $\frac{W}{P}$, which varies little, we find results such as are indicated in the following table.

Table of the Theoretical Values of $\frac{W}{P}$.

P	p	θ	$\frac{r}{T}+\frac{\theta}{2T-\theta}$	V	D	x	$\frac{DV}{10x}$	$\frac{W}{P}$
kg/cm^2	kg/cm^2	degrees		m/sec	kg/m^3			
10	5.8	22.258	1.0875	449.19	3.0570	0.9650	142.30	14.23
5	2.9	19.337	1.2028	439.85	1.5950	0.9650	72.70	14.54
2	1.16	16.310	1.3530	428.42	0.6738	0.9670	29.85	14.92
1	0.58	14.459	1.4639	420.04	0.3507	0.9697	15.19	15.19

The theoretical formula $\frac{W}{P}=15.20-0.96\log P$ gives the same figures as in the last volume to within about unity in the last order

In order to ascertain the relation between the value of this ratio and the pressure we may plot as abscissæ, not lengths proportional to P, but lengths proportional to log. P, and as ordinates the values of the ratio $\frac{W}{P}$ obtained from Regnault's tables. The points thus obtained range themselves almost exactly on a straight line from the value $P=1$ kilogramme per centimetre to $P=13$ kilogrammes per square centimetre. The digressions do not exceed one-tenth of one per cent, as will be seen by the curve on Plate II, where the points calculated from theory are marked by dots surrounded by circles. We can then write the following very simple empirical formula:

$$\frac{W}{P}=a-b\log P. \quad . \quad . \quad . \quad . \quad . \quad (7)$$

a and b being two constants for which the best values are $a=15.20$ and $b=0.96$, the discharge being given as before in grammes per second per square centimetre of the mouth of the nozzle.

This formula can also be replaced by the following, to which it is practically equivalent in practice $\left(\frac{b}{a}\right.$ being small):

$$W=aP'-\frac{b}{a}\log e, \quad . \quad . \quad . \quad . \quad . \quad (8)$$

e being the base of the natural logs, or in figures

$$W=15.20P^{0.9725}. \quad . \quad . \quad . \quad . \quad . \quad (9)$$

This is the form in which the formula is given by Grashof. (Grashof's formula is $W=15.26P^{0.9725}$ when the coefficient of discharge is equal to unity.[1])

In practice the first form is the most convenient and is the one I have recommended.

We will see that our experiments lead to slightly increasing the coefficient 15.20 and bring it to exactly the value 15.26 adopted by Grashof.

EXPERIMENTS.

7. Previous Experiments.—Previous to the work which I now publish and of which the complete results are given, we possessed several series of experiments on the flow of steam; especially in France those of Minary and Résal in 1861;[1] and more recently those of M. H. Parenty in 1891[2]; in America those of Napier in 1866[3]; then those of Peabody and Kunhard in 1888 and 1890,[4] and Miller and Read in 1895[5]; in England the experiments made quite recently by Rosenhain in 1899.[6] All these authors have used the same method, consisting in condensing the steam after it has passed through the nozzle under test in a surface condenser, and then weighing the water thus condensed during a certain lapse of time.

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[1] In certain German works a formula is given as Grashof's in which the coefficient is a little smaller than 15.26 (by about 1%) in order to allow for the contraction of the jet.

This method, which may be called the direct method, presents many inconveniences, however.

Firstly: One cannot operate on large nozzles without difficulty because a very large condenser is necessary for the experiments, and it is desirable to employ large nozzles—first, because then we approach the conditions of practice, and, second, the effect of friction and the thermal influence of the walls are reduced.

Secondly: The duration of a series of experiments is unavoidably long, since for each measurement it is necessary to wait until a considerable quantity of water has been condensed, which involves each trial lasting ten to fifteen minutes.

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[1] Annales des Mines, 5th série, Vol. IX.

[2] Annales de Chimie et de Physique, May 1896.

[3] On the Velocity of Steam and Other Gases. Spon, 1896.

[4] American Society of Mech. Eng., 1890.

[5] Technology Quarterly of Boston, 1895, Vol. VIII.

[6] Experiments on Steam-jets (Pro. of Civil Eng., 1900).

Thirdly: The method is not very precise because of the incessant variation of the steam-pressure. It is very difficult, in fact, to maintain the pressure of a boiler practically constant during some ten minutes, and the variations are reproduced at the nozzle. This involves taking averages, which complicates the experiment and renders it less precise.

Fourthly and finally: The entrained moisture is weighed in addition to the steam discharged, and consequently the measurement is falsified by a corresponding quantity nearly equal to the proportion of water, so that results obtained by this method are generally too large. For example, on the diagram in Plate II some of Résal's results are plotted (by triangles), and it will be seen that they are about 3 per cent too high. Those of Rosenhain and other authors give similar results.

8. Actual Experiments.—To avoid these sources of error all readings should be completed in a very short interval of time, as soon as the steady state of flow is properly established. The experiments can then be made rapidly and precisely. I achieved this result by the application of a new and indirect method consisting in condensing the steam discharged by the nozzle in a current of very cold water and at the same instant measuring the total discharge of water and the rise in its temperature. This method is founded on the use of an ejector condenser. The total quantity of water flowing after condensing the steam was ascertained by discharging the water through a conical nozzle under a head which could be accurately measured. Knowing the total quantity discharged Q and the rise of temperature, the weight of steam discharged X is deduced from the specific heat C and the total heat of formation of steam λ by the formula

$$X = Q\frac{C\theta}{\lambda - Ct_0}, \quad . \quad . \quad . \quad . \quad . \quad (10)$$

t_0 being the initial temperature of the water.

The advantages of this method are evident. The duration of each individual test did not exceed two to three minutes. The steady state was established in about a minute, and when all the instruments had reached a stable condition the reading of their indications was made in a few seconds. The conditions of pressure were then changed, and so on. The accuracy of the experiments I estimate as follows:

9. Estimation of the Precision of the Measurements. —The increase of temperature was on the average about 20° C., and the measurement of each of the initial and final temperatures in the ejector condenser were made to within one twentieth of a degree, which gave a mean error of 1 in 400. As I shall show later, however, a cause of error was always present which was quite important; namely, the displacement of the zero of the thermometer on the discharge side of the condenser during the course of the measurements. This displacement reached about two tenths of a degree, being nearly 1 per cent, or one half of 1 per cent, above or below the mean. The initial and final pressures were obtained by different gauges. At first metallic gauges were used and afterwards a mercury manometer provided with three-way cocks which allowed of the branches being placed in communication either with the atmosphere or with the interior of the pipes, when the pressure or the difference of pressures did not exceed 1 kilogramme per square centimetre. I was thus able to measure by this mercurial manometer either the initial or final pressures or their difference. The principal reading, that of the initial pressure P, was ordinarily made by means of a large Bourdon gauge especially constructed for my experiments and reading to $\frac{1}{100}$ of a kilogramme, which for 5 kilogrammes mean pressure corresponds to a mean approximation of 0.2 per cent. The mean head H on the nozzle for measuring the water discharged was about 450 millimetres. As the error of reading was less than $\frac{1}{2}$ millimetre, the error in calculating the discharge could not exceed an average of one in 1800. The coefficient of discharge of the nozzle was previously tested. The error which could enter into this determination, which will be described later on, was a few thousandths at the most.

The diameter of the steam-nozzles was obtained with great precision by means of mandrels calibrated to $\frac{1}{100}$ of a millimetre. As the mean diameter was 15 millimetres, the error in the diameter could attain 1 in 1500, or in the area 1 in 750.

A cause of error which was very difficult to eliminate was the expansion of the nozzles while the steam was passing through them. While the coefficient of expansion of the bronze was known to be 1 in 600, the temperature at the end of the nozzle could not be ascertained exactly, but it could be estimated certainly to within 40 degrees, and this was sufficiently close, because the resulting error was then less than 1 in 1500 in the diameter, or 1 in 750 in the area.

To sum up, then, all the individual errors of the different readings, that due to the displacement of the zero of the thermometer at the discharge excepted, are of the order of one to three one-thousandths. Added together they would give a total accidental error in the calculated results exceeding 1 per cent, but taking one with another, the mean error would be probably not more than a few thousandths.

Two sources of error must be examined which are inseparable from the method adopted: firstly, the entrained moisture, and, secondly, the effect of radiation from the apparatus.

Regarding the entrained water it should be remarked that our method avoided this cause of error as far as possible. Because if there were, for example, 1 per cent of entrained moisture, this quantity would tend to reduce slightly the velocity of discharge of the steam passing through the orifice, and consequently to diminish the total quantity flowing per second. But, on the other hand, some of the heat contained in this water will be given up to the water leaving the condenser and replace part of the heat lost by the reduction of the flow of steam; so that, finally, if the calculation be made, it is found that the actual deficit of heat will be only about half that corresponding to the proportion of water. This method, then, even if it does not eliminate the influence of the entrained water completely, reduces that influence by one half; and further, the error is in the opposite direction to that produced in the surface-condenser method. In place of adding to the discharge it decreases it.

But I arranged to avoid entrained moisture as far as possible, and also to measure its amount. A separator was placed in the steam-supply pipe just before the nozzle, and between this separator and the nozzle a sample of the steam was taken which was analyzed in an apparatus especially designed for this object, which permitted the moisture in the steam to be measured. In practice I never had more than two to three one-thousandths of water in the steam after passing the separator, so that the error in the measurements caused by the entrainment of water should be less than fifteen ten-thousandths.

As to the effect of radiation, though the apparatus was not lagged, this was relatively very small because of the large discharge of steam. In order to ascertain the importance of the loss of heat by radiation, I passed a small current of steam at atmospheric pressure through the apparatus and weighed the water condensed by the cooling action of the walls. The quantity thus obtained was 2.3 kilogrammes per hour. As the mean temperature of the parts exposed to radiation between the steam-nozzle and the discharge thermometer was generally much less than 100° C. during the experiments, it may be admitted that the radiation was less than 0.6 kilogrammes per hour, while the discharge was on the average 500 kilogrammes, or a loss of twelve ten-thousandths at the most. Finally, it may be observed also that the influence of radiation has the same effect as the entrained moisture and could cause only a deficit in the calculated quantity of steam, while we shall see that the experimental results give somewhat larger figures for the discharge than those calculated theoretically from Regnault's tables.

A difficulty arose in measuring the water discharged on account of the air contained in it. This air, at first dissolved in the water, is set free under the influence of the vacuum in the condenser and remains in this state up to the nozzle used for measuring the water, so that if precautions were not taken the discharge from this nozzle would be a mixture of water and air. Under these conditions measurements of the quantity of water discharged would have been erroneous. This difficulty was overcome as shown in Plate I by arranging a separator in the tank below the water-nozzle by means of which the air was disengaged and carried to the upper portion of the tank, where it was discharged through a small nozzle three millimetres in diameter, which permitted of its being measured. The water discharged by the principal nozzle was then quite clear and free from air.

10. Arrangement of the Apparatus. — The experiments were made during the winter of 1895–96 at St. Etienne, in the factory of Bietrix & Company, who obligingly placed at my disposal a boiler capable of giving 1000 kilogrammes of steam per hour at a pressure of 15 kilogrammes per square centimetre. The experiments were made in winter so as to be able to use cold water at a temperature of about 6° C. and obtain a rise of 40° to 50° C.

Three convergent nozzles were tested successively, then an orifice in a thin plate; finally we tested a convergent-divergent nozzle. The convergent nozzles are shown in Plate II. They had a final diameter of 10.49 millimetres, 16 millimetres, and 24.20 millimetres respectively. The thin-plate orifice, also shown in Plate II, had a diameter of 20.12 millimetres.

In order to make all the necessary readings the arrangement of apparatus was somewhat complicated. It is represented in perspective at the left and diagrammatically at the right of Plate I.

T is the nozzle under test arranged in the joint *I* between the steam-supply pipe *N* and the exhaust-pipe *B*.

The supply-pipe *N* had an inside diameter of 60 millimetres. The exhaust-pipe *D* (inside diameter 120 millimetres) was fitted with a plate *p* forming a screen for the current of steam leaving the nozzle so as to break up the jet.

A is the ejector condenser. The condensing water enters by the pipe *C* and leaves by the pipe *D*.

S is the water-nozzle of the condenser. The distance between it and the throat of the condenser could be varied by the handle *g*. By means of this nozzle the current of steam entering the condenser could be throttled.

K is a cock on the steam-supply pipe operated by the handle 18. This cock enabled the initial pressure at the nozzle to be varied at will, while the back pressure could be varied similarly by the condenser-nozzle *S*.

J is a centrifugal separator. The separated water is discharged by the cock 1 and the pipe 2.

L is the apparatus for measuring the moisture remaining in the steam. The description of this apparatus is given in Annales des Mines.[1] It is composed of a coil 4 heated by a flame 5, and a combining-chamber 6. The sample of steam obtained by the cock 3 is separated into two equal portions. One half goes directly to the combining-chamber, while the other is superheated in the coil 4. The mixture is discharged by the regulating angle-valve 9.

The superheat in the half passing through the coil 4 is measured by the thermometer t_3, and the superheat of the mixture by the thermometer t_2. The pressure in the chamber is observed by the gauge M_3. The moisture in the steam is obtained from the readings of these two thermometers and the gauge.

E is a sheet-iron tank in which the air is separated from the hot water leaving the condenser. This tank is divided about the middle by a semicircular baffle 13 under which the air collects. The bubbles rise to the upper part of the tank by the pipe 14, and the air flows into the atmosphere by the small nozzle *J*. The pressure of discharge of the air is measured by the water-gauge 19.

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[1] A. Rateau. Appareils servant à mesurer l'humidité d'une vapeur. Annales des Mines, April 1897.

F is the nozzle for measuring the discharge of water. It has a final area of 10.99 square centimetres at 15° C. The head under which the flow takes place is measured by the water-gauge *hh*. Adjacent to this gauge is a thermometer t_5 by which the temperature of the gauge could be measured and the density of the water deduced.

G is a barrel having a capacity of 350 litres into which the water is discharged by the nozzle and serves to calibrate the nozzle. The barrel is provided with a glass tube *KK*, allowing the exact level of the water to be ascertained. Further, a baffle 17, arranged in the arc of a circle, receives the jet of water in a thin film so that it falls into the barrel without shock and without disturbing the surface of the water.

M_1, M_2, M_3 are the pressure-gauges for measuring the initial and final pressures at the steam-nozzle *T*.

The mercury manometer is shown at *cd*. One of the branches *c* communicates with either the high-pressure side of the nozzle or the atmosphere by means of the three-way cock 11; the other branch *d* communicates with either the atmosphere or with the discharge side of the nozzle by the three-way cock 10. By means of these three-way cocks the initial or final pressures or the difference between them could be measured as the case required. In addition the barometric height and the temperature of the surrounding air were carefully observed so as to obtain the absolute value of the readings of the manometer.

The temperatures t_0 and t_1 of the current of water before and after passing through the condenser were measured by the thermometers *e* and *f*. These mercury thermometers were graduated to twentieths of a degree and were previously compared very carefully with a Baudin standard thermometer. I also reduced the scale to that of the air-thermometer, so that with the exception of the displacement of the zero, regarding which I will speak immediately, all temperatures are given to within one twentieth of a degree less, or one tenth of a degree more, than the scale of the air-thermometer.

11. **Remarks.**—Some observations are necessary on the subject of measuring the principal temperatures and pressures.

The reading of the pressure-gauges presented no difficulty. For pressures below 10 kilogrammes per square centimetre I employed the standard gauge M_1, and for pressures above this the standard gauge M_2. But the management of the mercury manometer offered some difficulties. The steam condensed in the tubes above the mercury and formed columns of water often broken up with air-bubbles, and the reading of the manometer had to be corrected for these broken columns of water. On account of this the readings of the mercury manometer were not always as exact as I could have wished. When the columns of water were not discontinuous, as sometimes happened, the correction could be made with certainty.

The thermometers *e* and *f* were plunged in small iron wells penetrating into the interior of the pipes conveying the water. These thermometer-cups were filled with mercury, so that the transmission of heat was very rapid. In fact the reading of the thermometers became steady in a few seconds. It may be asked whether these thermometers really indicated the true mean temperatures of the water. In the case of thermometer *e*, placed in the current of cold water, there can be no doubt that this was so, for the temperature of the cold water varied very little. But in the case of thermometer *f*, plunged in a current of water which was necessarily but little homogeneous, immediately at the outlet from the ejector, this was more doubtful, especially as the temperature of this water varied greatly from one experiment to another. I think, however, that the readings of these temperatures were always very close to the true mean, because on putting a thermometer into the water at the mouth of the nozzle *F*, where the temperature had evidently become uniform throughout the mass of water, I found that this thermometer always gave the same reading as thermometer *f* to within one tenth of a degree, sometimes more, sometimes less. This difference may be attributed to the walls of the tank *E* not immediately reaching their steady temperature after a change of conditions.

The thermometers used were made of ordinary glass. I had them carefully compared, as I said, with a standard Baudin thermometer to obtain the necessary corrections, if possible, to one twentieth of one degree. Unfortunately I found out too late that I had been wrong in not using a good thermometer made of hard glass for measuring the temperature of the hot water. I thought that up to 50° C. the displacement of the zero occasioned by the expansion of the glass could be neglected. I found that it was not so, however.

My first calculations having disclosed an error of more than 1 per cent between the experimental results and the theoretical figures, I suspected that the thermometer was incorrect, and standardized it again *de novo*. I found that its zero was displaced fully two tenths of a degree when it was subjected to a temperature of 50°, which was sometimes exceeded in my experiments, and that the zero returned to its original position when cold. This annoying variation in the zero was sufficient to render some of my results incorrect by over 1 per cent.

To reduce this influence to a minimum I ascertained the mean position of the zero due to the changes of temperature during several successive days. Finding that this mean differed by 0.15° from that which I had first obtained, I corrected all my results by this amount. For this reason the chart and the conclusions given here for the formula for the discharge are not identical with those in my Rapport sur les Turbines à Vapeur au Congrès de Mécanique Appliqué, 1890.[1]

12. **Calibration of the Water-nozzle.** — The determination of the total weight of water discharged depends upon the coefficient of discharge of the water-nozzle. Consequently it was necessary to measure this coefficient with the utmost possible accuracy at the outset. This calibration was repeated several times. The diameter of the mouth of the nozzle was 37.4 millimetres at 15° C.

Two glass reference-marks were fixed in the barrel into which the water was discharged by the nozzle, and the water contained in the barrel between these two points was carefully weighed. At an interval of two months the weights obtained were 278.2 kilogrammes and 278.05 kilogrammes at about 15° C.; so that the error to be allowed for in this measurement was not more than one in two thousand.

While the head H measured by the water-gauge hh remained quite constant, the time required to fill the barrel between these two marks was noted with a good seconds chronometer. To give an idea of the approximation obtained I produce below the calibration measurements made on the 27th and 30th of January, 1896. (See the Table below.)

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[1] This report was drawn up hurriedly a few days before the Congress. The portion relating to the flow of steam contains some errors which are rectified here.

EXAMPLES OF THE CALIBRATION OF THE WATER-NOZZLE.

Diameter of the nozzle 37.4 *mm. Contents of tank* 278.5 *litres.*

	Temperature t_1 of the Water.	Head, H.	Temperature t of the Water-manometer.	Time to Fill Tank.	Time reduced to $H=478$ mm.	Averages.	Coefficient of Discharge.
		Millim.		Seconds.	Seconds.	Seconds.	
Jan. 27, 1896	29.35°	549	22° C.	79.0	84.6	85.2	0.9711
	25.70	560	17	79.6	86.1		
	31.90	581	17	77.5	85.4		
	22.65	562	18	78.2	84.8		
Jan. 30, 1896	32	478	19	84.7	84.7	84.75	0.9763
	30	497	23	83.0	84.7		
	29	466	16	86.0	84.9		
	29	451	21	87.2	84.6		

As will be seen, the results differ by a few thousandths. Regarding the management of the chronometer, it will be understood that it was difficult to avoid errors of $\frac{1}{4}$ second which correspond to a relative error of three thousandths in a duration of about 80 seconds.

In calculating the discharge, I adopted the mean coefficient 0.9750, which is perhaps a little high. It is astonishing that the coefficient of discharge reached a value so close to unity; but that was the result of the experiments, and it was necessary to take the coefficient obtained.

I regret very much that I did not replace the 350-litre barrel by a much larger tank, say of 3 or 4 cubic metres capacity, so that the discharge could have been measured for some ten minutes. I would then have obtained an approximation certainly to one one-thousandth.

13. **Results of the Experiments.**—Three convergent nozzles were tested successively, and then an orifice in a thin plate. The nozzles and the orifice, which were of bronze, are represented to scale in Plate II. The convergent nozzles had, as I have already said, a diameter at the mouth of 10.49 millimetres, 15.19 millimetres, and 24.20 millimetres, respectively, at 15° C. The thin-plate orifice had a diameter of 20.12 millimetres at 15° C. These diameters were measured with great precision to 0.01 millimetre, and the orifices were previously corrected by passing calibrated mandrels through them so as to render them as perfectly circular as possible.

The results of the experiments and the calculations are given in the tables attached to this paper. The experiments are numbered 1 to 152; eleven have not been transcribed because they appear to me to contain very large accidental errors. These are numbered 9, 38, 56, 57, 58, 61, 82, 83, 84, 91, and 92; number 47 may also be omitted.

The first column of the tables gives the number of the experiment. Columns 2 and 3 are the absolute initial and back pressures. These figures are corrected from the observed readings and represent the true values. For example, a small quantity, about 0.01 kilogramme, due to the velocity of the steam in the supply-pipe, has been added to the initial pressure as indicated by the gauge. The gauge reading is due to the static pressure only, while the true initial pressure on the nozzle is the sum of the dynamic pressure and the static pressure.

Column 5 gives the head H on the axis of the water-nozzle,[1] and the figures correspond to the temperature of the water flowing through the nozzle.

Columns 6 and 7 indicate the temperatures t_0 and t_1 before and after the ejector condenser. As I have explained above, these are reduced to the readings of the air-thermometer. The difference between these figures gives the rise in temperature t_1-t_0 of the water (column 8).

The volume of water discharged was obtained by the formula

$$Q=KS\sqrt{2gH}, \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where Q = cubic centimetres per second,
S = area of the nozzle in square centimetres,
K = coefficient of discharge = 0.975,
H = head in centimetres.

The area S of the nozzle is corrected for expansion at the temperature of the current of hot water. The weight discharged in kilogrammes per second is written in column 9.

The quantity of steam flowing per second is then calculated by means of the total discharge of water and from the temperatures t_0 and t_1. Calling λ the total heat of the steam above 0° C., and C the mean specific heat of the water between the temperatures t_0 and t_1, X the weight of steam discharged per second by the nozzle, and $Q-X$ the quantity of injection water supplied to the condenser, the water resulting from the condensation of the steam being at the temperature t_1, then the quantity of heat given up by the steam[1] $H=X(\lambda-Ct_1)$, while the quantity of water $Q-X$ is heated from t_0 to t_1 and absorbs $(Q-X)C(t_1-t_0)$ heat units. We have then the following equation:

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[1] This was a brass nozzle.

$$X(\lambda-Ct_1)=(Q-X)C(t_1-t_0),$$

whence

$$X=Q\frac{C\theta}{\lambda-Ct_0}, \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where $\theta=t_1-t_0$.

The quantity of heat λ is given by the formula or by Regnault's Tables, and the specific heat C of the water is given by the well-known expression

$$C=1+0.4\frac{t}{10^3}-0.9\frac{t^2}{10^6}.$$

The steam discharged, calculated in this manner, expressed in grammes per second, is given in column 10 of the tables.

Dividing by the area of the nozzle, we have the discharge per square centimetre in column 11.

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[1] During the expansion of the steam in the nozzle a certain quantity of heat is transformed into mechanical energy, but this quantity is immediately and entirely given back again by the destruction of the kinetic energy of the jet, so that it is quite correct to take $X(\lambda-Ct_1)$ as the quantity of heat given up by the steam.

A remark is necessary here. What value should be taken for the area of a nozzle? The diameter I have given is measured cold (15° C.), while under the action of the current of steam the nozzle becomes heated and expands. It is impossible to know exactly what to take for the temperature of the walls, because the current of steam has a varying pressure and also a varying temperature during its passage through the nozzle; and further, the walls are subjected to different temperatures at different points. It seems to me to be useless to try to calculate the true temperature of the nozzle in each experiment.

I adopted finally a mean temperature of 120° C. and based the calculations on the diameters corresponding to this temperature. This is one of the difficulties of these researches on the flow of steam and also on the flow of gases. To eliminate this cause of error entirely, it would be advantageous to employ metals having a very small coefficient of expansion, such as the new alloys of iron and nickel. When my experiments were made, however, these alloys were not yet known. It is unnecessary to give too much importance to this cause of error. The variation of temperature of the nozzle did not differ by more than about 40° C. from the mean temperature adopted (120° C.), and for bronze 40° corresponds to a linear expansion of one in fifteen hundred, so that the error in the area did not exceed 1 in 750, or but little more than one tenth of one per cent.

Finally, the last column of the table gives the ratio of the discharge of steam W to the initial pressure P

when p is less than $0.58P$; or the ratio of the actual discharge W to the maximum discharge W_m when the ratio $\frac{p}{P}$ is greater than 0.58. These two cases are always distinguished in the tables, because in the first case the discharge depends on the initial pressure P which is constant (as is p), while in the second case it depends on the back pressure also.

We will now examine the results of the experiments and compare them with those given by theory.

14. Convergent Nozzles.—The ratio $\frac{W}{P}$ varies but slightly. It oscillates about the value 15. A very clear chart can be made by plotting as ordinates the value of this ratio, as is done on Plate II, where I have taken as abscissæ the logarithm of the initial pressure. This graphic representation is very satisfactory because the origin of the coordinates is a long way off the paper, so that variations of only 1 per cent correspond to a relatively large height. To give an idea of the amplification of the ordinates, I have drawn the lines ab and $a'b'$ corresponding to a difference of 1 per cent above and below the theoretical line AB. It will be seen that the majority of the points lie above the theoretic line AB. Only a few points are more than 2 per cent away from this line. If the mean distance of the experimental points from the theoretical curve be taken for each nozzle, the following results are found:

For the nozzle 10.49 millimetres diameter (21 experiments) the mean departure is 1.17 per cent high.

For the nozzle 15.19 millimetres diameter (19 experiments) the mean departure is 0.69 per cent high.

For the nozzle 24.20 millimetres diameter (19 experiments) the mean departure is 0.25 per cent high.

These are very small differences and clearly show the satisfactory agreement between the experiments and theory.

If the curve for the mean results of the experiments be plotted, it will be seen that it follows the direction of the theoretical curve closely, but is a few thousandths above it. This difference between experiment and theory can be accounted for by several causes.

First of all, in the theoretical calculations I have taken the mechanical equivalent of the calorie as 425. This is a little low, as is now recognized. Now $E^{\frac{1}{2}}$ enters into the expression for the speed of the steam. If 428 be taken instead of 425, the figures for the theoretical discharge will be increased by 0.35 per cent, and the mean difference between experiment and theory will not be more than 0.35 per cent.

Further, we have seen that the calibration of the water-nozzle was not as accurate as could have been wished; perhaps a certain error is to be feared there. I do not believe that it exceeds 0.3 per cent.

Finally there was a displacement of the zero of the thermometer measuring the temperature of the hot water. This is certainly the most important source of error. I stated above that I was led to modify the first standardization by 0.15° after having carefully observed the variations of the thermometer. Nevertheless it may be that the mean position of the zero during the experiments was slightly different from that which I finally adopted. An error of less than one tenth of a degree would be sufficient to account for the remaining error of 0.35 per cent.

I have indicated on the chart the points corresponding to the old experiments by Minary and Résal. These points are surrounded by a triangle; one of them has not been plotted, as it is outside the limits of the figure. It appears that these experiments gave very much larger results than ours, which is explained, as I have already remarked, by the method employed not allowing of a correction being made for the effect of the entrained moisture. The error reaches 5.2 per cent and averages 2.5 per cent.

Similarly Rosenhain's experiments on nozzle No. 4 give an error reaching 3 to 4 per cent.

We shall see shortly in analyzing Hirn's experiments on the flow of air that he had the same trouble with that fluid.

Thus there is a satisfactory agreement between experiment and theory, but the experimental discharge appears to be a little larger than what we should expect from theory.

Should the theoretical formula be followed in practice, or should we adopt the coefficients given by the preceding experiments? If it be desired to use the theoretical formula, the coefficient must be taken as 15.20

for $E=425$, or 15.25 for $E=428$; and if it be desired to follow experiment, a value of 15.32 at most should be taken for the coefficient. The practical formula then will be

$$\frac{W}{P}=15.20 \text{ to } 15.32-0.96 \log P, \quad . \quad . \quad (13)$$

within the limits of our experiments; that is to say, between $P=1$ and $P=12$ kg. per square centimetre.

In my report to the Congrès de Mécanique Appliqué I gave 15.42, which resulted from a summary glance over the experiments, as the first term of the second member of this equation, and for the second term $-\log P$. At the International Congress in Glasgow I suggested the value 15.20 resulting from the theoretical formula in which E is taken as 425. These are the limiting values of this term. I estimate that the most probable actual value is 15.26 with a possible error less than 0.4 per cent. This is exactly the coefficient in Grashof's formula which was deduced from theory by the first method given on page 15. Grashof's formula, however, gives a too rapid decrease for the ratio $\frac{W}{P}$, as will be seen from the figure on Plate II, where the line CD corresponds to this formula.

The value of the coefficient of $\log P$ is between 0.96 and unity. At first I always took unity because of the simplification which results; nevertheless it is better to take 0.96, which is probably more exact because it is given by the thermodynamic calculation.

15. **Remarks.**—The agreement between experiment and theory to less than 1 per cent leads to the conclusion that, contrary to the opinion sometimes stated, there is no sensible retardation in the condensation of the steam. In fact the calculation takes into account the condensation during the expansion, and the proportion of steam condensed when the pressure has become equal to $0.58P$ is about 0.3 to 0.36 per cent, depending on the value of P. Now if any retardation in the condensation occurred, experiment would reveal a deficit amounting to 0.3 per cent to 0.35 per cent in the quantity of steam discharged. But in place of a deficit, a too high value is always found so that it may be concluded that if there is any retardation of the condensation it is very small, of the order of 0.00001 of a second.

On the other hand it will be seen also that the coefficient of discharge for convergent nozzles of the form tested by me is very close to unity for values of p less than $0.58P$, since the experimental discharge is always found to be larger than the theoretical. A similar conclusion results from Hirn's experiments on air. When p is greater than $0.58P$ this is not the case, however.

Finally, if, conversely, we suppose *a priori* that the coefficient of discharge is unity, then the value of E, the mechanical equivalent of the calorie which will make the experimental discharge equal to the theoretical, is found to be 431, which differs by 0.7 per cent only from the value 428 generally adopted.

16. Experiments with p larger than $0.58P$. — When the back pressure is larger than $0.58P$ the flow depends not only on the initial pressure P, but also on the back pressure p. The results can be best represented by taking the ratio of the actual discharge W to the maximum discharge W_m, which occurs when the back pressure is less than $0.58P$. This ratio $\frac{W}{W_m}$ is then practically independent of the ratio $\frac{p}{P}$ of the pressures. The points can be plotted on cross-section paper and the curve of $\frac{W}{W_m}$ as a function of $\frac{p}{P}$ constructed.

For the experiments where p was larger than $0.58P$ the tables give, in column 12, the theoretical maximum discharge W_m calculated by the formula given above:

$$W_m = P(15.26 - 0.96 \log P), \quad . \quad . \quad . \quad (14)$$

and in column 13 the ratio $\frac{W}{W_m}$.

The results of these experiments have been plotted in this manner on Plate III, the results for each of the three convergent nozzles experimented on being differently indicated.

It will be seen that the points fall very well on a curve of the elliptic form, as was pointed out by M. H. Parenty.[1] As I have remarked above, the larger departures from this curve may be considered as being due to the great uncertainty that sometimes occurred in measuring the difference of the pressures by the mercury manometer when the column of water surmounting the column of mercury was interrupted by bubbles of air. The standard gauges also gave but a poor approximation at low pressures. It will be noticed further that the greatest differences occur in the results for the largest nozzle 24.20 millimetres in diameter. The results given by the small nozzle are much more regular.

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[1] Annales de Chimie et de Physique, May 1896.

To compare the results obtained with theory the curve *AB*, based on the thermodynamic calculation, has been drawn. This curve is easily obtained by taking the ratios of the function $\frac{DV}{x}$ defined on page 9 to the maximum value of this function. It is simply necessary to take the reciprocal of the ratios of the numbers in column 8 of the table on page 11 to the smallest of these numbers, 0.70276.

It will be noticed that the experimental curve lies a little below the theoretical curve, as would be expected, and that it becomes horizontal a little later than the theoretical curve, which means that the maximum experimental discharge is not attained until the back pressure is lower than $0.58P$. By taking the ratio between the ordinates of the experimental and theoretical curves the coefficient of discharge for the converging nozzles is obtained, from which it will be seen that this coefficient starts with a value about 0.94

when the ratio of the pressures is in the neighborhood of unity and gradually increases towards the value unity. As the experimental curve and also the theoretical curve are very nearly ellipses, the ratio $\frac{W}{W_m}$ as a function of the ratio $\frac{p}{P}$ of the pressures can be expressed by a very simple formula, but it does not seem worth while to go into this.

We shall see that the curves for air are also very nearly quadrants of an ellipse.

17. Orifice in a Thin Plate.—The orifice in a thin plate does not behave in the same way as the convergent nozzles. The discharge does not reach a maximum for p equal to or a little less than $0.58P$, but increases constantly as p falls.

The experimental figures are given in the tables either with the ratio $\frac{W}{P}$ or, which is better, the ratio $\frac{W}{W_m}$.

The results relating to this orifice are also plotted graphically on Plate III up to the value of $\frac{p}{P}$ equal to 0.4. It will be noticed that the points fall very regularly on a curve EF which is always rising. If the ratio between the ordinates of this curve to those of the theoretical curve AB be taken, it will be seen that the coefficient of discharge for the thin-plate orifice starts at a value of 0.61 for small differences of pressure and increases very regularly to the value 0.87 which is reached for very small back pressures. But the curve of this coefficient of discharge as a function of the ratio $\frac{p}{P}$ is not at all a straight line. It shows a slight depression around $\frac{p}{P}=0.7$. On the contrary, if the ratio of the ordinates of the curve for the thin-plate orifice be taken to those of the experimental curve *CD* for convergent nozzles instead of to the theoretical curve, it is found that the ratio increases in a straight line. On Plate III points representing this ratio for certain values of $\frac{p}{P}$ are marked. It will be seen that these points be very nearly on a straight line *GH*, which of course is tangent to the thin-plate orifice curve at the point *A*, which is on the ordinate of the point where the curve *CD* becomes horizontal.

COMPARISON WITH THE RESULTS OF HIRN'S EXPERIMENTS ON AIR.

In 1885 Hirn made some very precise experiments on the flow of air through nozzles and orifices in thin plates. It is interesting to compare his results with those of our experiments on steam. Hirn used the direct method. After passing through a holder the air was caused to flow into a receptacle in which a vacuum had previously been formed. By a certain method of registering he measured the fall of the holder at equal intervals of time, as well as the initial and discharge pressures.

On analyzing his numerical results, given in the Annales de Chimie et de Physique, March 1886, it will be noted, first of all, that Hirn did not take into account in his calculations the slight contraction of the orifice occasioned by the lowering of the temperature due to the expansion of the air. It is easy to make the correction, however. Again, it appears to me to be wrong to dry the air after it left the holder and before passing the nozzle, because dry air was flowing through the nozzle, while the gas-holder measured humid air. As the volumetric discharge of gases (referred to the initial pressure and temperature) is the same for all gases at the same temperature, and that of the vapor could not differ very much, it would have been better, in my opinion, if he had not dried the air. The resulting error would only have been a fraction of the proportion of water vapor. What is the error to be expected from this cause in Hirn's experiments? At 10° C., the mean temperature during the experiments, the vapor pressure of water is 0.0125 kilogramme per square centimeter, and its density is equal to 0.00195. The error due to this cause would increase the results by about 2 per cent. Hirn's experiments were made on two convergent nozzles, one having 9° and the other 13° convergence; also on a conico-cylindrical nozzle, and on two thin-plate orifices. The experiments on the nozzles clearly showed that the discharge becomes a maximum when the back pressure falls slightly below the theoretical figure for permanent gases ($0.526P$). Taking the discharge per unit area of the mouth of the orifice, we have the following figures: 19.80 litres per square centimetre per second for the 9° nozzle at the initial temperature of 15.75° C.; 19.77 litres per square centimetre per second for the 13° nozzle, at the temperature of 8° C.; 18.85 litres per square centimetre for the conico-cylindrical nozzle at the temperature of 6.5° C. The theoretical maximum discharge for these nozzles, calculated by the formula $Q=1.164T$, where T is the absolute initial temperature and Q is expressed in litres per square centimetre per second, is respectively 19.77, 19.50, and 19.47 litres.

For the conico-cylindrical nozzle, then, the experimental discharge is less than the theoretical, which is to be expected because of the losses by friction in the cylindrical part. But for the convergent nozzles the experimental discharge is larger than the theoretical by about 0.75 per cent on the average. Here we have the same results that we have already observed for the vapor of water. It should be noted always that this exaggeration of the measured discharge could arise, as I have explained above, from Hirn's having dried the air.

I have plotted on Plate IV the experimental points, taking the ratio of the pressures as abscissæ and the ratio of the observed discharge to the maximum discharge as ordinates. I have also plotted a few theoretical points.

It will be seen that for the nozzle having a convergence of 13° and for the conico-cylindrical nozzle the experimental curve AB is very close to the theoretical curve except in the neighborhood of the maximum discharge. But for the nozzle with 9° convergence the experimental curve *CD* departs considerably from the theoretical curve, particularly towards the origin. It would seem that there was some systematic error in the series of measurements relative to this nozzle.

The results obtained by the two thin-plate orifices lie very nicely along the curve *EF* on Plate IV. Taking the ratio between the ordinates of this curve and those of the curve *AB* for convergent nozzles, we find figures which increase along a straight line in the same way as was obtained in the case of steam. That is to say, the points representing these ratios which are plotted on the chart lie very exactly on a straight line *GH* tangent to the curve *EF* for the thin-plate orifice. The origin of this line is 0.629, while Hirn found by direct experiment that for small differences of pressures the coefficient of discharge for thin-plate orifices was 0.633.

Comparing the curves thus obtained for the flow of air with those obtained for the flow of steam, it will be noticed that they are entirely analogous and can be almost superposed. There is always this difference, that the theoretical maximum of the curves for air occurs at $\frac{p}{P}=0.526$, while for steam it occurs at $\frac{p}{P}=0.58$. It will be noticed also that the straight line which represents the ratio of the coefficient of discharge for thin-plate orifices to that for convergent nozzles for steam lies about 2 per cent above the corresponding line for air. I am unable to decide whether this difference is due to experimental errors or to the different nature of the fluids.

1.—*January* 30, 1896. *Outside temperature* 16° *C.* *Barometer* 736 *mm.* =0.997 *kg.* *Convergent nozzle B.*
Diam. 15.9 *mm.* *Area at* 120° *C.*: 1.819 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, t_1-t_0	Total Water Discharged, kgs/sec	Steam Discharged.		Ratio, $\frac{W}{P}$
	Initial, P kgs/cm^2	Final, p kgs/cm^2			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
1	9.26	0.144	0.0155	553	5.40	50.50	45.10	3.491	240.97	132.48	14.311
2	8.53	0.157	0.0184	589	5.50	46.05	40.55	3.609	224.37	123.36	14.467
3	8.43	0.144	0.0171	582	5.40	45.65	40.25	3.599	222.05	122.08	14.487
4	7.49	0.144	0.0192	590	5.50	41.05	35.55	3.620	197.72	108.69	14.520
5	6.68	0.144	0.0216	591.5	5.50	37.40	31.90	3.629	178.20	97.97	14.678
6	5.52	0.103	0.0187	584.5	6.82	33.35	26.53	3.611	148.27	81.51	14.786
7	4.48	0.0895	0.0200	575.5	6.72	28.60	21.88	3.588	121.50	67.01	14.902
8	3.64	0.103	0.0283	579	6.07	23.65	17.58	3.604	98.61	54.22	14.890
10	4.14	0.166	0.2816	473	5.67	27.65	21.98	3.253	111.02	61.03	14.737
11	4.10	1.945	0.4744	426	5.59	28.50	22.91	3.087	109.79	60.19	14.717

3.—*February* 4, 1896. *Outside temperature* 16° *C.* *Barometer* 734 *mm.* =0.996 *kg.* *Convergent nozzle C.* *Diam.* 24.20 *mm.* *Area at* 120° *C.*: 4.6176 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1=t_0$	Total Water Discharged, kgs/sec	Steam Discharged.		Ratio $\frac{W}{P}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
25	2.44	0.154	0.0631	564	11.25	41.31	30.06	3.538	168.05	36.395	14.92
26	1.99	0.133	0.0668	560	10.85	35.80	24.95	3.532	139.59	30.229	15.19
27	1.53	0.108	0.0706	564	10.65	29.73	19.08	3.551	107.69	23.319	15.24
28	1.15	0.127	0.1103	551	10.40	24.78	14.38	3.514	80.63	17.461	15.19
29	1.22	0.122	0.1000	578	6.90	22.10	15.20	3.601	86.76	18.788	15.40
30	1.80	0.198	0.1100	572	6.77	29.10	22.33	3.577	125.86	27.256	15.14
31	2.93	0.171	0.0584	574.5	6.57	42.38	35.81	3.570	199.99	43.311	14.78
32	3.57	0.184	0.0516	550.5	6.47	50.98	44.51	3.482	241.64	52.352	14.66
33	4.03	0.235	0.0547	570.5	6.12	55.02	48.90	3.537	269.21	58.298	14.46
34	1.00	0.355	0.3550	558.5	6.07	18.52	12.55	3.542	70.20	15.206	15.21
35	3.05	1.462	0.4800	391	5.70	50.52	44.82	2.935	205.86	44.579	14.61
36	2.97	0.964	0.3223	452	5.77	46.62	40.85	3.160	202.05	43.757	14.73
37	2.95	1.685	0.5710	364	5.62	50.21	44.59	2.833	198.77	43.047	14.59

5.—*February* 20, 1896. *Outside temperature* 19° *C.* *Barometer* 732 *mm.* =0.991 *kg.* *Convergent nozzle A.*
Diam. 10.49 *mm.* *Area at* 120° *C.*: 0.8686 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, t_1-t_0	Total Water Discharged, kgs/sec	Steam Discharged.		Ratio, $\frac{W}{P}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
79	9.16	0.231	0.0252	583	7.77	28.61	20.84	3.612	115.57	133.34	14.522
80	8.42	0.201	0.0239	581.7	8.21	27.43	19.22	3.608	106.78	122.73	14.583
81	7.76	0.191	0.0246	572.6	8.46	26.35	17.89	3.581	98.74	113.68	14.653
85	10.74	0.151	0.0140	599	6.42	30.37	23.95	3.659	133.82	154.06	14.339
86	10.41	0.151	0.0145	594	6.49	29.88	23.39	3.644	130.25	149.96	14.401
87	9.98	0.151	0.0151	589.5	6.71	29.20	22.49	3.631	124.93	143.83	14.407
88	9.48	0.141	0.0149	586.5	6.82	28.31	21.49	3.622	119.23	137.27	14.475
89	8.99	0.141	0.0157	583.3	6.71	27.26	20.55	3.613	113.83	131.05	14.572
90	8.46	0.131	0.0155	567	6.62	26.21	19.59	3.563	107.11	123.32	14.572
93	11.34	0.191	0.0168	565.5	6.19	31.81	25.62	3.554	138.84	159.84	14.098
94	11.42	0.191	0.0167	580.5	6.26	32.05	25.79	3.600	141.58	163.00	14.268
95	11.44	0.191	0.0167	580.5	6.29	32.25	25.96	3.600	142.51	164.07	14.338
96	11.14	0.151	0.0136	574.5	6.27	31.56	25.29	3.582	138.21	159.12	14.339
97	10.46	0.151	0.0144	576.5	6.27	30.43	24.16	3.589	132.46	152.50	14.574
98	11.84	0.151	0.0128	570	6.24	33.30	27.06	3.566	147.04	169.29	14.294
99	11.54	0.151	0.0131	565	6.12	33.05	26.93	3.551	145.75	167.81	14.536

6.—*March* 10, 1896. *Outside temperature* 20° *C.* *Barometer* 732 *mm.* =0.992 *kg.* *Convergent nozzle A.* *Diam.* 10.49 *mm.* *Area at* 120° *C.*: 0.6886 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged,		Ratio, $\frac{W}{P}$
	Initial, P kgs/cm^2	Final, p kgs/cm^2			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
100	10.74	4.89	0.4553	352.5	5.92	37.00	31.08	2.801	132.86	152.96	14.236
101	10.54	5.24	0.4767	353.5	5.97	36.55	30.58	2.805	130.98	150.80	14.30
102	10.49	5.49	0.5234	353.5	5.97	36.50	30.53	2.805	130.78	150.56	14.35
103	10.46	5.74	0.5488	352.5	5.97	36.45	30.48	2.801	130.39	150.12	14.34
104	10.43	5.99	0.5744	351.5	5.97	36.35	30.38	2.797	129.79	149.43	14.32

4.—*February* 5, 1896. *Outside temperature* 17° *C.* *Barometer* 735 *mm.* =0.996 *kg.* *Convergent nozzle C.*
Diam. 24.20 *mm.* *Area at* 120° *C.:* 4.6176 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.		Ratio, $\frac{W}{P}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
55	1.19	0.644	0.5412	522	6.37	21.70	15.33	3.423	82.48	17.985	15.11
62	1.58	0.646	0.4089	532	5.47	25.62	20.15	3.452	109.47	23.784	15.03
63	1.59	0.713	0.4485	519	5.47	25.97	20.50	3.409	109.48	23.887	15.03
64	1.60	0.796	0.4975	505	5.47	26.55	21.08	3.362	111.65	24.191	15.13
65	1.63	0.887	0.5442	490	5.49	26.96	21.47	3.311	111.52	24.293	14.89
67	1.64	0.717	0.4372	503	5.47	27.08	21.61	3.355	114.18	24.788	15.10

7.—*March* 12, 1896. *Outside temperature* 21° *C.* *Barometer* 729 *mm.* =0.988 *kg.* *Thin-plate orifice D.* *Diam.* 20.12 *mm.* *Area at* 120° *C.*: 3.1918 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.		Ratio, $\frac{W}{P}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
117	4.04	1.820	0.4505	577	6.22	35.95	29.73	3.585	165.63	51.924	12.85
118	4.02	1.534	0.3816	548	6.12	36·35	30.23	3.493	164.20	51.444	12.80
119	3.96	1.236	0.3121	489	6.04	37.55	31.51	3.298	161.63	50.640	12.79
120	3.86	0.993	0.2573	445	6.02	38.27	32.25	3.145	157.83	49.448	12.82
121	2.41	0.298	0.1237	571.3	6.12	24.09	17.97	3.578	100.77	31.573	13.10
122	2.54	0.728	0.2866	506.5	6.12	25.87	19.05	3.368	104.12	32.626	12.85
123	2.57	0.958	0.3728	482.5	5.97	26.35	20.38	3.287	104.84	32.846	12.78
124	2.61	1.148	0.4398	460.6	6.12	26.80	20.68	3.211	103.94	32.566	12.48
125	2.27	1.168	0.5145	464.2	6.12	23.45	17.33	3.226	87.69	27.480	12.11
126	2.36	1.378	0.5839	453.2	6.12	23.45	17.33	3.187	86.58	27.130	11.50

8.—*March* 13, 1896. *Outside temperature* 23° *C.* *Barometer* 727 *mm.* =0.985 *kg.* *Thin-plate orifice D.*
Diameter 20.12 *mm.* *Area at* 120° *C.:* 3.1918 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Milli-metres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.		Ratio, $\frac{W}{P}$
	Initial, P kgs/cm	Final, p kgs/cm^2			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
129	4.135	1.355	0.3285	428	5.87	40.03	34.16	3.083	163.65	51.275	12.40
130	4.135	1.635	0.3955	396	5.82	41.08	35.26	2.965	162.42	50.910	12.31
131	3.835	1.885	0.4920	387	5.84	37.68	31.84	2.934	145.38	45.551	11.88
132	3.805	2.045	0.5375	377	5.87	37.32	31.45	2.897	141.75	44.421	11.68
133	2.985	1.595	0.5344	421	5.82	29.25	23.43	3.068	112.27	35.182	11.73

9.—*March* 17, 1896. *Outside temperature* 16° *C.* *Barometer* 730 *mm.* = 0.989 *kg.* *Convergent nozzle B.* *Diameter* 15.19 *mm.* *Area at* 120° *C.:* 1.819 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.		Ratio, $\frac{W}{P}$
	Initial, P kgs/cm^2	Final, p kgs/cm^2			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12
144	5.266	0.369	0.0701	585.2	6.12	30.97	24.85	3.616	139.03	76.523	14.52
145	3.893	0.369	0.0948	585	6.12	24.50	18.38	3.621	103.50	56.896	14.61
146	2.912	0.229	0.0787	571.5	6.17	20.35	14.18	3.582	79.34	43.690	15.60
147	5.395	0.249	0.0462	593.5	5.97	31.13	25.16	3.642	141.67	77.881	14.44
148	4.444	0.209	0.0470	583	5.97	26.95	20.98	3.613	117.58	64.640	14.55
149	3.423	0.169	0.0494	578.2	5.97	22.30	16.33	3.601	91.52	50.371	14.72
150	5.495	0.179	0.0326	579.7	5.90	31.97	26.07	3.598	144.98	79.698	14.50
151	4.474	0.169	0.0378	569	5.92	27.34	21.42	3.569	118.57	65.176	14.56

1.—*January* 30, 1896. *Outside temperature* 16° *C.* *Barometer* 736 *mm.* =0.997 *kg.* *Convergent nozzle B.* *Diameter* 15.19 *mm.* *Area at* 120° *C.:* 1.819 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.			Ratio, $\frac{W}{W_m}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	Theoretic Maximum, W_m gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12	13
12	3.84	2.933	0.764	451	5.60	23.50	17.90	3.179	88.44	48.62	56.43	0.816
13	3.92	3.492	0.891	482	5.60	18.70	13.10	3.290	66.94	36.75	57.57	0.638
14	4.04	3.716	0.920	220	5.41	22.85	17.45	2.222	60.29	33.09	59.29	0.558
15	5.00	4.826	0.965	131	5.07	23.10	18.03	1.714	47.76	26.26	72.95	0.360

2.—*January* 31, 1896. *Outside temperature* 17° *C. Barometer* 735 *mm.* =0.996 *kg. Convergent nozzle D. Diameter* 15.19 *mm. Area at* 120° *C.:* 1.819 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.			Ratio, $\frac{W}{W_m}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total gr/sec	Per sq. cm., W gr/sec	Theoretic Maximum, W_m gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12	13
16	4.23	3.603	0.852	458.5	5.60	22.10	16.50	3.207	82.09	45.12	61.99	0.728
17	5.42	5.164	0.953	498	5.72	17.72	12.00	3.345	62.01	34.08	78.89	0.432
18	5.52	4.581	0.830	399	5.60	29.98	24.38	2.986	112.46	61.82	80.30	0.770
19	5.75	5.113	0.892	413	5.62	26.50	20.88	3.041	98.00	53.87	83.50	0.645
20	5.56	5.132	0.923	443.5	5.62	22.30	16.68	3.154	81.23	44.66	80.87	0.552
21	6.24	5.763	0.924	415.5	5.57	24.78	19.21	3.051	90.32	49.65	90.45	0.549
22	6.43	6.236	0.970	485	5.62	17.33	11.71	3.301	59.42	32.72	93.11	0.351
23	6.46	6.296	0.975	120.4	5.17	27.00	21.83	1.641	55.12	30.30	93.53	0.324
24	6.05	5.496	0.908	390.5	5.54	26.45	20.91	2.954	95.23	52.35	87.79	0.597

3.—*February* 4, 1896. *Outside temperature* 16° *C. Barometer* 734 *mm.* =0.995 *kg. Convergent nozzle C. Diameter* 24.20 *mm. Area at* 120° *C.:* 4.6176 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Milli-metres.	Temperatures.		Rise of Temper-ature, $t_1 - t_0$	Total Water Dis-charged, kgs/sec	Steam Discharged.			Ratio, $\frac{W}{Wm}$
	Initial, P kgs/cm^2	Final, p kgs/cm^2			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	Theoretic Maximum, Wm gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12	13
39	2.72	1.943	0.744	373	5.52	41.97	36.45	2.876	162.97	35.51	40.33	0.881
40	3.23	2.025	0.853	348	5.47	41.77	36.50	2.777	157.20	34.04	47.67	0.714
41	3.02	2.754	0.901	387.5	5.57	32.11	26.54	2.941	121.83	26.38	45.65	0.591
42	2.85	2.731	0.948	437.5	5.57	23.00	17.43	3.132	85.26	18.46	42.20	0.437
43	4.05	2.702	0.951	370.5	5.52	31.76	26.24	2.876	117.22	25.38	59.43	0.427
44	5.05	4.958	0.982	414.7	5.52	23.79	18.27	3.049	86.16	18.66	73.66	0.253
45	5.50	5.458	0.992	453	5.57	17.82	12.25	3.190	60.33	13.06	80.02	0.159
46	2.95	5.850	0.966	478	5.32	19.21	13.89	3.276	70.98	15.37	43.57	0.353
48	4.03	3.964	0.984	472	5.32	18.03	13.34	3.252	67.38	14.59	59.15	0.247
49	4.75	4.656	0.980	414.5	5.32	25.18	19.86	3.047	93.67	20.29	69.39	0.292
50	4.96	4.922	0.992	151	5.07	24.14	19.07	1.830	53.94	11.68	72.38	0.161
51	5.02	4.987	0.993	104	4.92	26.35	21.43	1.526	50.53	10.94	72.23	0.149
52	1.95	1.717	0.881	440	5.32	23.40	18.08	3.141	89.15	19.31	29.21	0.661
53	1.58	1.236	0.782	457	5.32	23.55	18.23	3.201	91.90	19.90	23.78	0.837
54	1.32	1.138	0.862	484.3	5.32	17.57	12.25	3.308	64.00	13.86	19.96	0.694

4.—*February* 5, 1896. *Outside temperature* 17° *C.* *Barometer* 735 *mm.* =0.996 *kg.* *Convergent nozzle C.* *Diameter* 24.20 *mm.* *Area at* 120° *C.*: 4.6176 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.			Ratio, $\frac{W}{W_m}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	Theoretic Maximum, W_m gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12	13
56	1.21	0.787	0.650	521.5	5.42	20.50	15.08	3.416	81.46	17.64	18.34	0.961
57	1.24	0.871	0.702	513.5	5.42	20.30	14.88	3.395	79.68	17.25	18.78	0.918
58	1.07	0.750	0.701	521.5	5.47	18.62	13.15	3.423	71.29	15.44	16.29	0.948
59	1.18	0.937	0.794	513.5	5.47	17.97	12.50	3.396	67.16	14.54	17.90	0.812
60	1.16	1.003	0.865	517.0	5.47	15.83	10.36	3.409	55.86	12.10	17.61	0.687
66	1.64	0.966	0.590	477	5.47	27.39	21.92	3.267	112.72	24.42	24.66	0.991
68	1.68	1.077	0.641	462	5.42	27.59	22.17	3.215	112.20	24.30	25.24	0.963
69	1.50	1.036	0.687	466	5.42	24.68	19.26	3.261	98.11	21.25	22.60	0.940
70	1.72	1.366	0.794	442	5.42	25.08	19.66	3.146	97.22	21.07	25.83	0.816
71	2.16	1.924	0.891	421	5.42	24.78	19.36	3.072	92.78	20.09	32.25	0.623
72	5.94	5.689	0.958	305.5	5.02	45.87	40.85	2.590	163.19	35.34	86.24	0.410
73	6.05	5.855	0.968	335	5.02	39.88	34.86	2.727	146.54	31.64	87.81	0.360
74	6.14	5.995	0.977	357.3	5.02	34.25	29.23	2.823	127.11	27.53	89.06	0.310
75	6.26	6.157	0.983	389.7	5.20	28.90	23.70	2.953	106.77	23.34	90.74	0.257
76	6.40	6.330	0.989	432	5.22	23.69	18.47	3.112	88.47	19.16	92.71	0.207
77	6.58	6.535	0.993	481	5.22	19.36	14.14	3.287	71.48	15.48	95.24	0.162
78	6.67	6.639	0.995	496.5	5.20	16.83	11.63	3.340	59.71	12.93	96.51	0.134

6.—*March* 10, 1896. *Outside temperature* 20° *C.* *Barometer* 732 *mm.* =0.992 *kg.* *Convergent nozzle A.* *Diameter* 10.49 *mm.* *Area at* 120° *C.:* 0.8686 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, t_1-t_0	Total Water Discharged, kgs/sec	Steam Discharged.			Ratio, $\frac{W}{W_m}$
	Initial, P kgs/cm²	Final, p kgs/cm²			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	Theoretic Maximum, W_m gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12	13
105	10.31	6.10	0.600	344.3	5.90	36.00	30.10	2.769	127.30	146.54	147.30	0.995
106	10.29	6.44	0.626	344.3	5.90	35.75	29.85	2.765	126.07	145.14	147.02	0.987
107	10.21	6.69	0.655	346.3	5.88	34.75	28.87	2.778	122.50	141.03	145.91	0.966
108	10.17	6.94	0.683	349	5.88	34.05	28.17	2.789	120.05	138.21	145.37	0.951
109	10.14	7.19	0.710	356	5.88	33.05	27.17	2.818	117.00	134.69	144.90	0.929
110	10.09	7.44	0.738	364	5.88	31.76	25.88	2.851	112.73	129.78	144.25	0.899
111	10.19	7.74	0.760	368	5.88	31.22	25.34	2.867	110.97	127.76	145.65	0.877
112	10.44	7.99	0.766	366	5.88	31.67	25.79	2.859	112.56	129.60	149.10	0.869
113	10.52	8.44	0.803	380.7	5.88	29.79	23.91	2.917	106.47	122.58	150.21	0.816
114	10.56	8.94	0.846	403.5	5.88	27.25	21.37	3.005	98.01	112.84	150.76	0.754
115	10.67	9.49	0.890	428	5.88	23.99	18.11	3.097	85.58	98.51	152.29	0.647
116	10.78	9.92	0.921	439	5.88	21.70	15.82	3.138	75.72	87.16	153.81	0.567

7.—*March* 12, 1896. *Outside temperature* 21° *C*. *Barometer* 729 *mm*. =0.988 *kg*. *Thin-plate orifice D*. *Diameter* 20.12 *mm*. *Area at* 120° *C*.: 3.1918 *sq. cm*.

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.			Ratio, $\frac{W}{W_m}$
	Initial, P kgs/cm^2	Final, p kgs/cm^2			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	Theoretic Maximum, W_m gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12	13
117	4.04	0.156	0.0390	577	6.22	35.95	29.73	3.585	165.72	51.92	59.29	0.876
118	4.02	0.441	0.1096	548	6.12	36.35	30.23	3.493	164.19	51.44	59.01	0.872
119	3.96	0.738	0.1863	489	6.04	37.55	31.51	3.298	161.64	50.64	58.15	0.871
120	3.86	0.993	0.2573	445	6.02	38.27	32.25	3.145	157.82	49.45	56.72	0.872
121	2.41	0.298	0.1237	571.3	6.12	24.09	17.97	3.578	100.77	31.56	35.86	0.884
122	2.54	0.728	0.2866	506.5	6.12	25.87	19.05	3.368	104.12	32.63	37.73	0.865
123	2.57	0.958	0.3728	482.5	5.97	26.35	20.38	3.287	104.84	32.75	38.16	0.858
124	2.61	1.148	0.4398	460.6	6.12	26.80	20.68	3.211	103.93	32.57	38.74	0.841
125	2.27	1.168	0.5145	464.2	6.12	23.45	17.33	3.226	87.69	27.44	33.84	0.811
126	2.36	1.378	0.5839	453.2	6.12	23.45	17.33	3.187	86.58	27.13	35.13	0.772
127	2.51	1.668	0.665	439.2	6.12	23.55	17.43	3.145	85.64	26.84	37.30	0.720
128	2.24	1.738	0.776	459.8	6.12	19.22	13.10	3.218	66.03	20.69	33.40	0.619

8.—*March* 13, 1896. *Outside temperature* 23° *C.* *Barometer* 727 *mm.* =0.985 *kg.* *Thin-plate orifice D.* *Diameter* 20.12 *mm.* *Area at* 120° *C.*: 3.1918 *sq. cm.*

No.	Absolute Pressures.		Ratio, $\frac{p}{P}$	Head, H Millimetres.	Temperatures.		Rise of Temperature, $t_1 - t_0$	Total Water Discharged, kgs/sec	Steam Discharged.			Ratio, $\frac{W}{W_m}$
	Initial, P kgs/cm^2	Final, p kgs/cm^2			Entering, t_0	Leaving, t_1			Total, gr/sec	Per sq. cm., W gr/sec	Theoretic Maximum, W_m gr/sec	
1	2	3	4	5	6	7	8	9	10	11	12	13
129	4.135	1.355	0.329	428	5.87	40.03	34.16	3.083	163.54	51.27	60.63	0.846
130	4.135	1.635	0.396	396	5.82	41.08	35.26	2.965	162.43	50.91	60.63	0.840
131	3.835	1.885	0.492	397	5.84	37.68	31.84	2.934	145.38	45.55	56.40	0.808
132	3.805	2.045	0.538	377	5.87	37.32	31.45	2.897	141.75	44.42	55.97	0.794
133	2.985	1.595	0.534	421	5.82	29.25	23.43	3.068	112.27	35.18	44.15	0.797
134	3.035	1.805	0.595	409.6	5.82	29.10	23.28	3.026	109.92	34.45	44.87	0.768
135	3.185	2.045	0.642	399.6	5.82	29.20	23.38	2.989	109.01	34.15	47.03	0.726
136	2.635	1.678	0.637	423	5.82	25.18	19.36	3.077	93.18	29.16	39.26	0.743
137	2.985	2.349	0.786	456.5	5.67	22.65	16.98	3.199	84.88	26.57	44.15	0.602
138	3.235	2.711	0.838	455.5	5.67	21.95	16.28	3.196	81.09	25.41	47.75	0.532
139	3.485	3.036	0.870	450.5	5.72	21.50	15.78	3.181	78.14	24.48	51.36	0.477
140	3.885	3.516	0.905	453.5	5.72	20.90	15.18	3.190	75.23	23.57	57.13	0.413
141	4.385	4.113	0.936	454.3	5.70	19.61	19.91	3.193	68.88	21.58	64.20	0.336
142	4.635	4.409	0.950	462.2	5.70	18.71	13.01	3.222	64.93	20.34	67.76	0.300
143	4.985	4.814	0.965	480	5.70	17.02	11.32	3.284	57.50	18.02	72.74	0.248

NOTE ON THE FLOW OF HOT WATER THROUGH NOZZLES.

In this short note I propose to analyze the phenomenon of the flow of hot water through a convergent nozzle and to explain the results obtained by Sauvage & Pulin in experiments made by them in 1892,[1] which appear very singular at the first glance.

The theory developed in the preceding work relating to the flow of steam can be extended to the somewhat more complex problem of hot water which is partially vaporized during the discharge. The most simple case, which I will discuss first, is that in which the water in the high-pressure receptacle is initially at exactly the temperature T_0 corresponding to the pressure P_0 of the steam; that is to say, just on the point of evaporating. It will then begin to evaporate as soon as its pressure and temperature fall. A remarkable circumstance gives to this phenomenon a very special interest, namely, that the quantity vaporized on the one hand and the velocity of flow of the mixture of water and steam formed on the other hand are both practically proportional to the fall of temperature. Considering the entropy diagram,[1] Fig. 3, let *AD* be the curve of the entropy of water, and *EF* that of saturated steam. Let the point *A* represent the state of the hot water

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[1] Ed. Sauvage, Ecoulement de l'eau des Chaudières (Annales des Mines, 9th Series, Vol. II, page 192).

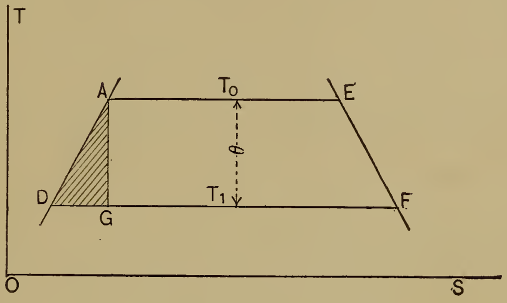

Fig. 3.

at the temperature T_0 and the corresponding pressure P_0.

During the expansion in the nozzle a small quantity of steam is generated which forms a more or less homogeneous mixture with the remaining water. It may be admitted that the expansion is adiabatic on account of the very short time during which the fluid remains in contact with the walls of the nozzle. This adiabatic expansion is represented on the entropy diagram by the line *AG* parallel to the axis of temperatures. When the mixture has reached the temperature T_1 differing from T_0 by the quantity θ, the proportion of steam

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[1] The utilization of the entropy diagram for the vapor of water was made the subject of some remarkable articles by Prof. Boulvin in the Revue de Mécanique in 1897 and 1901.

generated is given by the ratio of the segment of the straight line *DG* to *DF*.[1] Now the curve *AD* of the entropy of water is practically the same as its tangent at the point *A*, if the fall of temperature is not large. *DG* is then proportional to the fall in temperature θ, with a tendency, however, to increase a little quicker than θ, and one can write that the ratio of *DG* to *DF* is very nearly proportional to θ; that is to say,

$$x=a\theta,$$

a being a coefficient which depends on the initial temperature T_0. The value of this coefficient can be easily ascertained if the entropy diagram be drawn to a large scale.

On the other hand the velocity of flow V at the moment when the temperature has fallen to T_1 is given by the area of the triangle *AGD*. The base of this triangle is approximately equal to $x\frac{r}{T_0}$, r being the heat of vaporization of the water. We have then the following relation:

$$\frac{V^2}{2g}=\frac{1}{2}Ex\theta\frac{r}{T_0}=\frac{1}{2}Ea\theta^2\frac{r}{T_0},$$

whence

$$V=\theta\sqrt{gEa\frac{r}{T_0}}. \quad . \quad . \quad . \quad . \quad . \quad (1)$$

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[1] We may safely assume that there is no retardation of the vaporization so that the quantity of steam in the mixture is that corresponding to the segment *DG* at each instant.

We will now calculate the area of nozzle necessary to 7 allow unit mass of water to flow when the temperature has fallen by θ°. This area is equal to the specific volume v of the mixture of water and steam divided by the velocity V.

The specific volume v is equal to $1-x+\frac{x}{D}$, D being the density of saturated steam at the temperature $T_1=T_0-\theta$, or replacing x by its value $a\theta$, we have

$$v=1-a\theta+\frac{a\theta}{D}. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Dividing v by V, we have the area S:

$$S=\frac{1}{\sqrt{gEa\frac{r}{T_0}}}\left(\frac{1}{\theta}+\frac{a}{D}-a\right). \quad . \quad . \quad . \quad . \quad (3)$$

For any given initial temperature a and r are fixed. In the expression for S, consequently, the only variables are the fall of temperature θ and the density D which is a function of $T_0-\theta$; consequently S increases or decreases in proportion to the quantity $\frac{1}{\theta}+\frac{a}{D}$, and since a is very small, S is practically proportional to this quantity. Now when θ increases $\frac{1}{\theta}$ decreases. But on the other hand the density D diminishes and consequently the second term $\frac{a}{D}$ increases, so that a time arrives when the increase of the second term compensates for the decrease of the first, and the quantity $\frac{1}{\theta}+\frac{a}{D}$ then passes through a minimum.

In order to ascertain the position of the minimum it is necessary to express D as a function of the temperature.

In default of a simple relation between these quantities the problem can be solved in any particular case by the aid of Regnault's tables; for example:

Suppose the initial pressure P equals 10 kilogrammes per square centimetre. The corresponding temperature T_0 equals 178.886+273°. By the entropy diagram the value of a is found to be 0.00216; the density is given by the tables. It should be noted that D must be expressed in kilogrammes per cubic decimetre.

The following are the values of the quantity $\frac{1}{\theta}+\frac{a}{D}$ for increasing values of θ in the neighborhood of the minimum:

For $\theta=$	6° C.	$\frac{1}{\theta}+\frac{a}{D}=0.6501$
	7°	0.6374
	8°	0.6300
	9°	0.6274
	10°	0.6293
	11°	0.6325
	12°	0.6355

Plotting the curve, it is seen that the minimum occurs for $\theta=9.2°$. The absolute pressure corresponding to this value of θ is 8.044 kilogrammes per square centi-

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metre. Consequently when the pressure falls below about 80 per cent of the initial pressure it is necessary for the nozzle to diverge in order that the mixture of water and steam should continue to expand.

If the nozzle is simply convergent, the pressure at the mouth of the nozzle will be exactly equal to the value just obtained (i.e., 8.044 kg.). If the discharge takes place into a space in which the pressure is much less than this, as, for example, into the atmosphere, a sudden expansion occurs which is much stronger than in the case of the discharge of steam alone. The jet is seen to swell suddenly and take the form of a very wide paraboloid, as is well shown by the photographs reproduced in M. Sauvage's memoir.

The lateral expansion of the jet is enormous at the mouth of the nozzle because, as we shall see, the speed of flow of the mixture is not high and it is proportionately greater as the pressure in the exhaust space is lower than the pressure p giving the maximum output. The discharge per unit of area of the mouth of the nozzle can be calculated from the pressure p which makes the quantity $\frac{1}{\theta}+\frac{a}{D}$ a minimum. Making this calculation for the case under consideration, we find first that the quantity of steam generated x is equal to 1.99 per cent; that the specific volume of the mixture reaches the value 5.78; that is to say, at the mouth of the nozzle the steam occupies a volume 4.78 times that of the water; that the speed of the mixture at the mouth is 28.55 metres per second; and finally that the weight of fluid discharged is 494 grammes per square centimetre of orifice per second.

It is interesting to compare this last figure with that obtained if the nozzle was discharging initially saturated steam. In that case the quantity discharged would be 143 grammes per square centimetre per second. In the case of the hot water, therefore, the weight discharged is 3.5 times larger.

If the expansion is to be continued in the nozzle, the throat must be followed by a divergent portion. In Fig. 4 the curve *CMD* of the areas as ordinates is given as a function of the fall of temperature for the case where P_0 is 6 kilogrammes per square centimetre, the water being initially at the temperature of vaporization at this pressure.

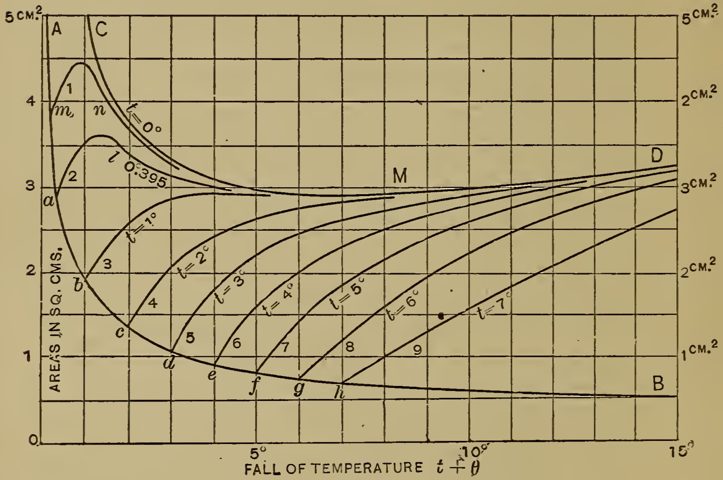

FIG. 4.

In the case of the flow of initially saturated steam, it was found that the value of the ratio $\frac{p}{P}$ corresponding to the minimum area (that is to say also, to the maximum discharge) is equal to 0.58, and that it is sensibly independent of the initial pressure P_0. In the case of hot water we have found, when P_0 equals 10 kilogrammes per square centimetre, that the value of the ratio corresponding to the minimum area is equal to 0.804, a value very much larger than the preceding. On repeating the same calculations for different initial pressures it is found that the value of the ratio $\frac{p}{P}$ corresponding to the minimum varies considerably. It approaches unity as the pressure P_0 decreases; thus for $P_0=6$ kilogrammes per square centimetre we find that the fall of temperature θ, which gives the minimum value for $\frac{1}{\theta}-\frac{a}{D}$, is 7.15 degrees, whence $\frac{p}{P}=0.829$; the speed of flow at the mouth of the nozzle is now only 22.5 metres per second, and the discharge 344 grammes per second.

Let it be supposed now that the water is initially at a temperature lower by t^0 than T_0, the temperature of vaporization at the pressure P_0. As long as the fall of pressure is insufficient to cause evaporation the water will remain in the liquid state and the discharge will follow the laws for liquids. When the pressure falls below this limit, however, partial evaporation will take place, and again a point will be reached at which the increase of the specific volume will compensate the increase of velocity. At this moment the area of the flow will become a minimum. The decrement of temperature t has the effect, then, of rapidly increasing the possible discharge per unit section of a given nozzle. We shall see later, by an example, that, except for very small values of t, the minimum area corresponds exactly to the point where evaporation begins to occur.

Calling P' the pressure existing at the point where evaporation begins (namely, the pressure corresponding to the temperature T_0-t for saturated steam) and θ the fall of temperature from this point, the speed of flow is given by the relation

$$\frac{V^2}{2g}=\frac{P_0-P'}{D}+\frac{1}{2}gEa\theta^2\frac{r}{T_0-t}, \quad . \quad . \quad . \quad (4)$$

D being the density of the liquid and a having the signification already given. The specific volume of the mixture is always given by the expression

$$V=1-a\theta+\frac{a\theta}{D}.$$

The analytical discussion in this case is even more difficult than in the preceding; but any particular case can be solved by means of Regnault's tables. It will be seen from the above that only a slight fall of temperature is sufficient to increase the discharge enormously. Thus, for example, to obtain a discharge of 494 grammes per second for a pressure of 10 kilogrammes per square centimetre, the speed must increase to 4.94 metres per second. For this speed

$$P_0 - P' = 0.124, \quad \text{whence} \quad t = 0.54^\circ.$$

This low value clearly shows the considerable influence which the decrement of temperature has on the discharge from a convergent nozzle.

Now in different parts of a boiler the temperature of the water is always a little lower than the temperature of vaporization. This difference will easily amount to several degrees, and is very variable, not only from point to point in the boiler, but also from one moment to another, depending on the state of the internal currents and the activity of the evaporation. One would consequently expect to find extremely variable discharges, such as have been shown by the experiments of Sauvage & Pulin. The increase of the discharge with reference to that taking place if there were no decrement of temperature, forms an indication of the amount of the decrement in temperature.

As this decrement of temperature increases, the flow tends to approach more and more that of a simple liquid and does not differ at all when the initial temperature of the water $T_0 - t$ is equal to or less than that of the vapor corresponding to the pressure P_1 in the exhaust space. The discharge will then reach the value

$$\sqrt{2g \cdot \frac{P_0 - P_1}{D}}$$

per square centimetre per second. The actual discharge is always found to lie between this last value and that which we have previously calculated for the case when the water is initially at the temperature T_0. For example, for an initial pressure of 6 kilogrammes per square centimetre and discharging into the atmosphere, the discharge per second per square centimetre should lie between 3130 grammes and 344 grammes, the figure just given. In their experiments Sauvage & Pulin found discharges for a pressure of 6 kilogrammes per square centimetre, varying irregularly around the value 1350 grammes, which would appear to indicate that under the conditions of the experiments the water was at a temperature about 6 degrees lower than that of steam at the same pressure.

For the sake of example I have given in Fig. 4 nozzle areas for the discharge of hot water from an initial pressure of 6 kilogrammes per square centimetre with differences of temperature increasing by single degrees. The total fall of temperature $(t+\theta)$ is plotted as abscissæ, while the ordinates indicate the area of flow in square centimetres for a discharge of 1 kilogramme per second. The curve AB represents the case of cold water (without evaporation); CMD that of hot water initially at the temperature of vaporization (where $t=0$); curves 3, 4, . . . 9 correspond to the cases where the decrement of temperature is 1°, 2°, etc.

For a decrement of 2°, for example, the water at first flows without causing evaporation and the areas decrease up to the point C; then, because evaporation takes place, the area increases, from which it will be seen that if the nozzle is convergent hot water will be discharged without any portion being vaporized, at its initial temperature and at the pressure corresponding to that temperature for saturated steam.

This is not quite general, however. If the curve of areas be drawn for decrements of temperature less than about 1 degree it will be found that two successive minima are obtained, as, for example, curve 2. The first minimum a is an angular point; the other occurs in the neighborhood of a point M on the curve CD. Further, when t is less than 0.395 degree, the first minimum is larger than the second, as will be seen from curve 1. It would appear from the singular form of this curve of areas that the flow in a convergent nozzle ought to undergo a discontinuity because it is impossible for the fluid to be in the states corresponding to the part of the curve between the points m and n. Since the areas of the nozzle are always decreasing, there must be a sudden jump from m to n; that is to say, an instantaneous partial evaporation, and, which is very curious, an equally instantaneous increase in the velocity.

This remarkable phenomenon of a sudden variation in the evaporation and in the speed demands very careful examination. It would appear incompatible *a priori* with the laws of inertia of material bodies; but on thinking of the matter one can see, I think, that these discontinuities are not impossible when fluid bodies are being experimented with, which can be decomposed actually and not only theoretically into very small elementary masses.

I will conclude with these remarks, which are sufficient to show that the flow of a hot liquid presents peculiarities very well worthy of attention.

APPENDIX.

To those wishing to study this subject further, the following bibliography may be of assistance:

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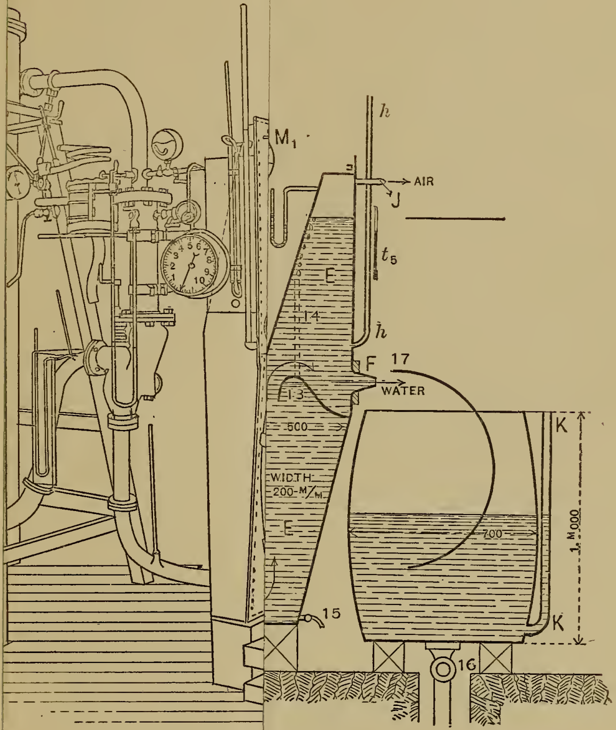

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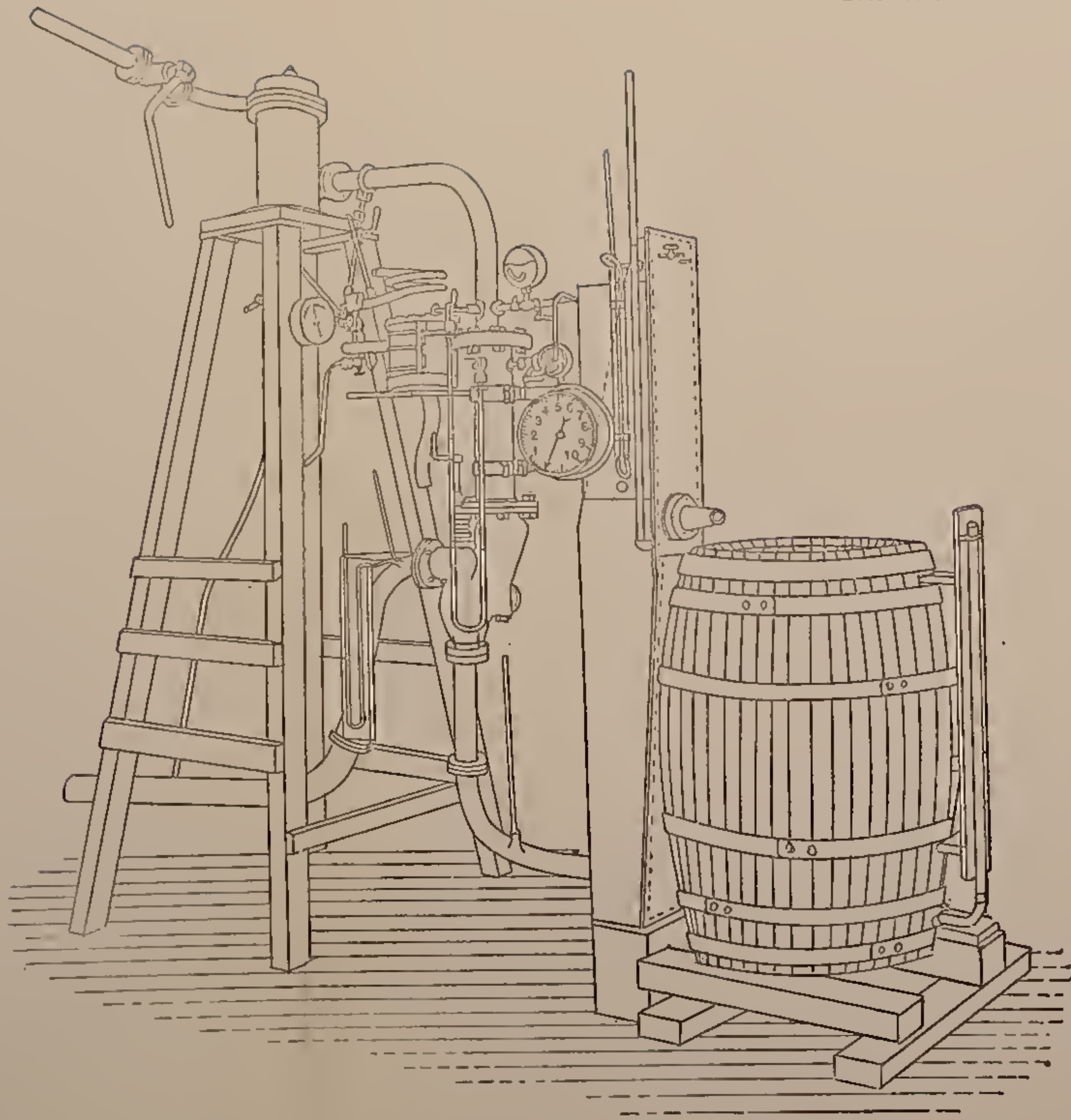

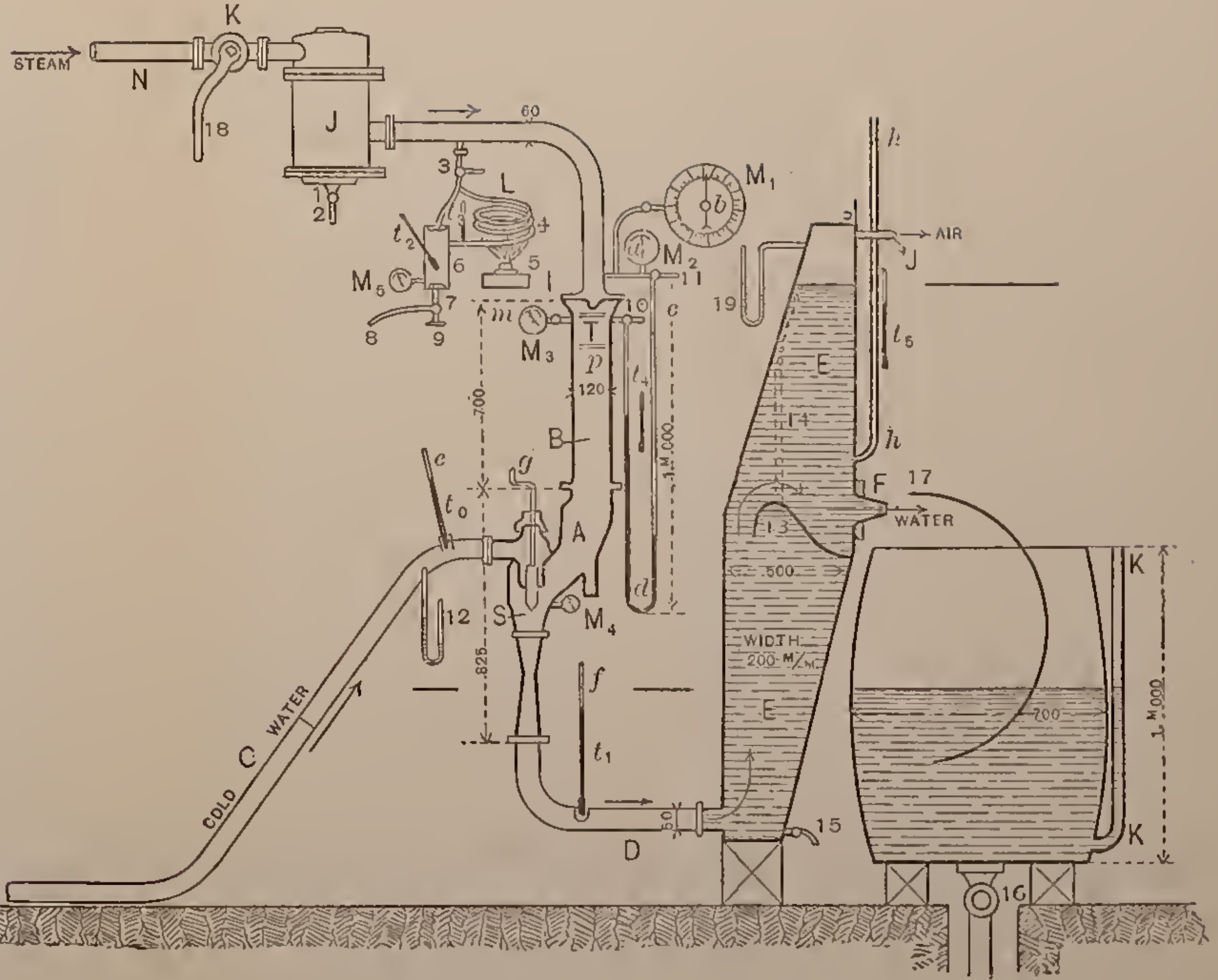

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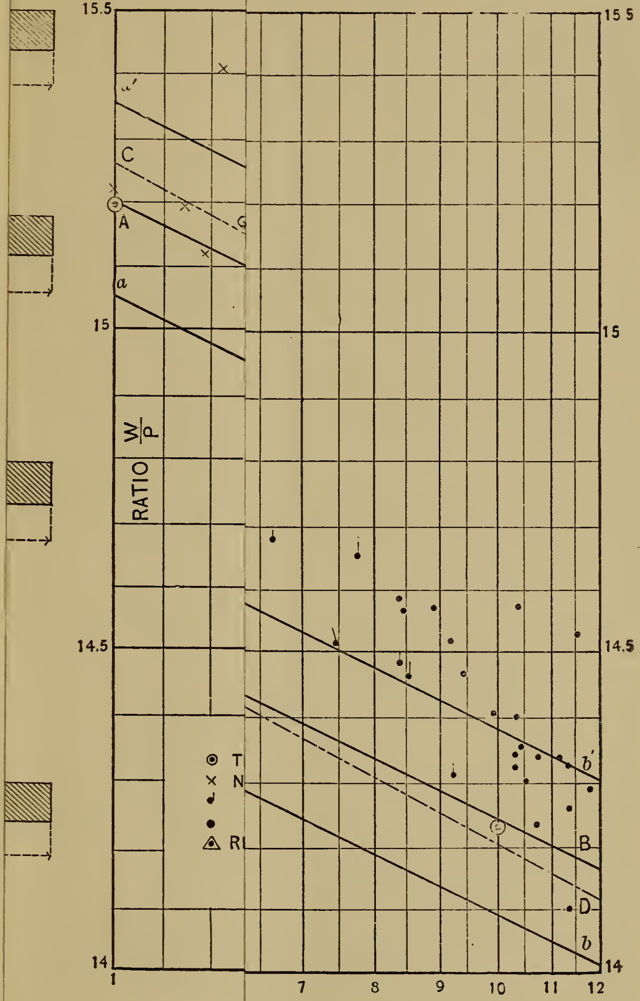
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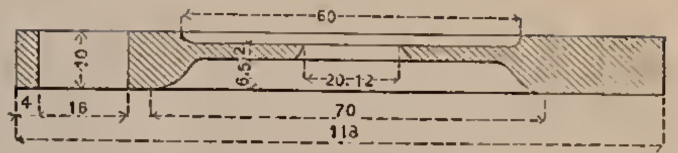

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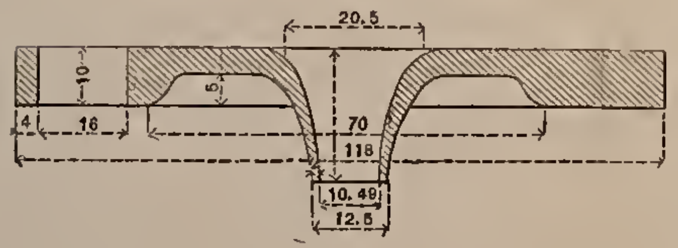

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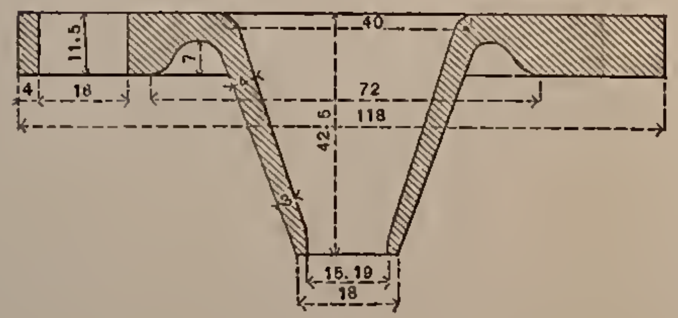

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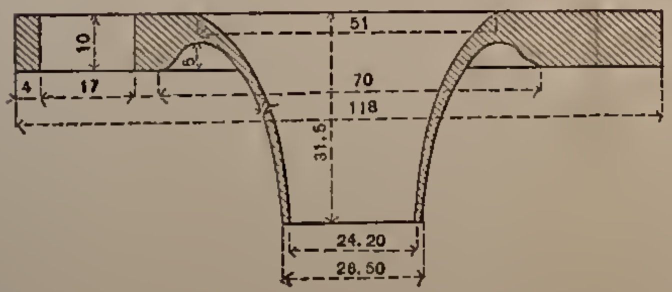

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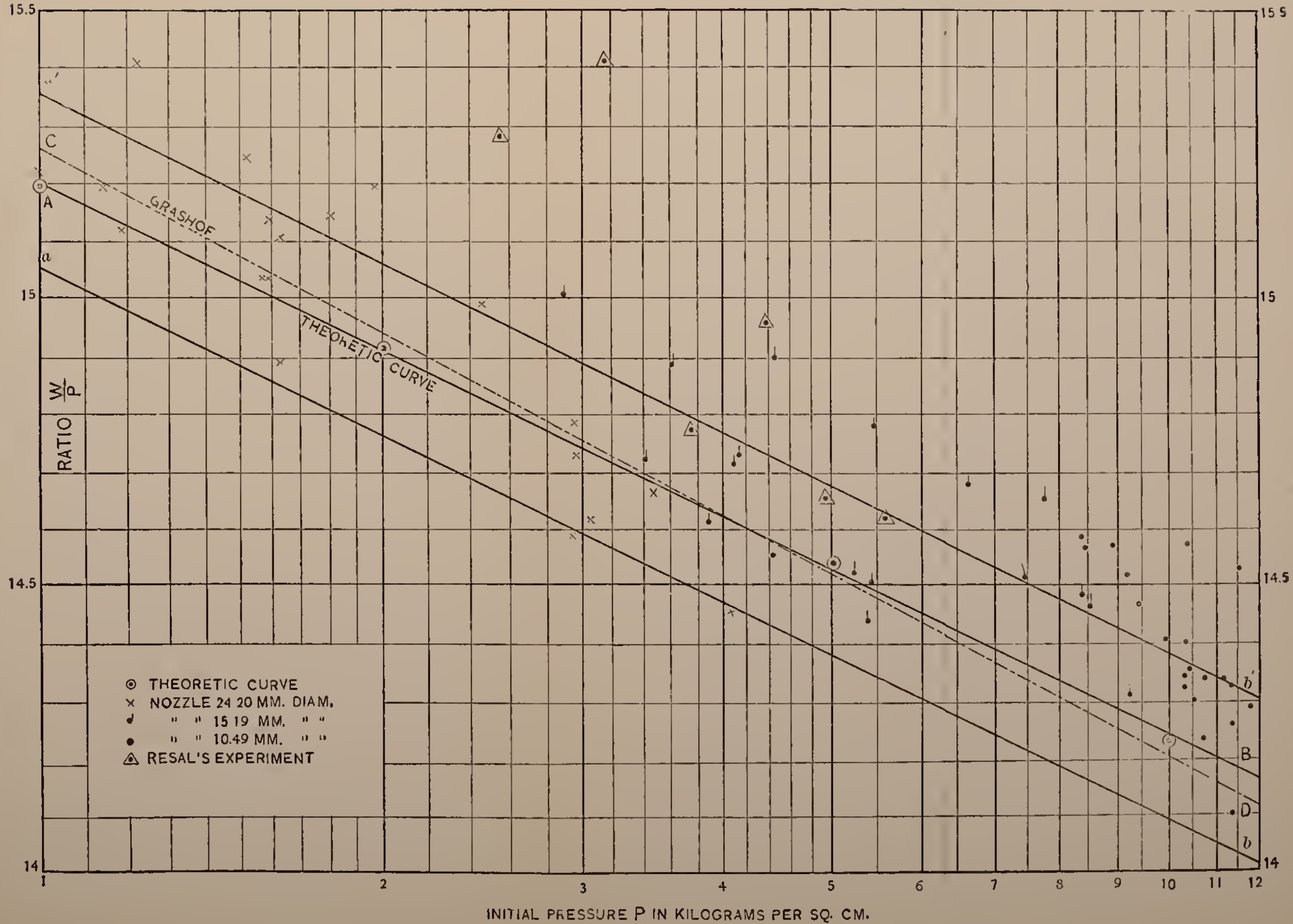

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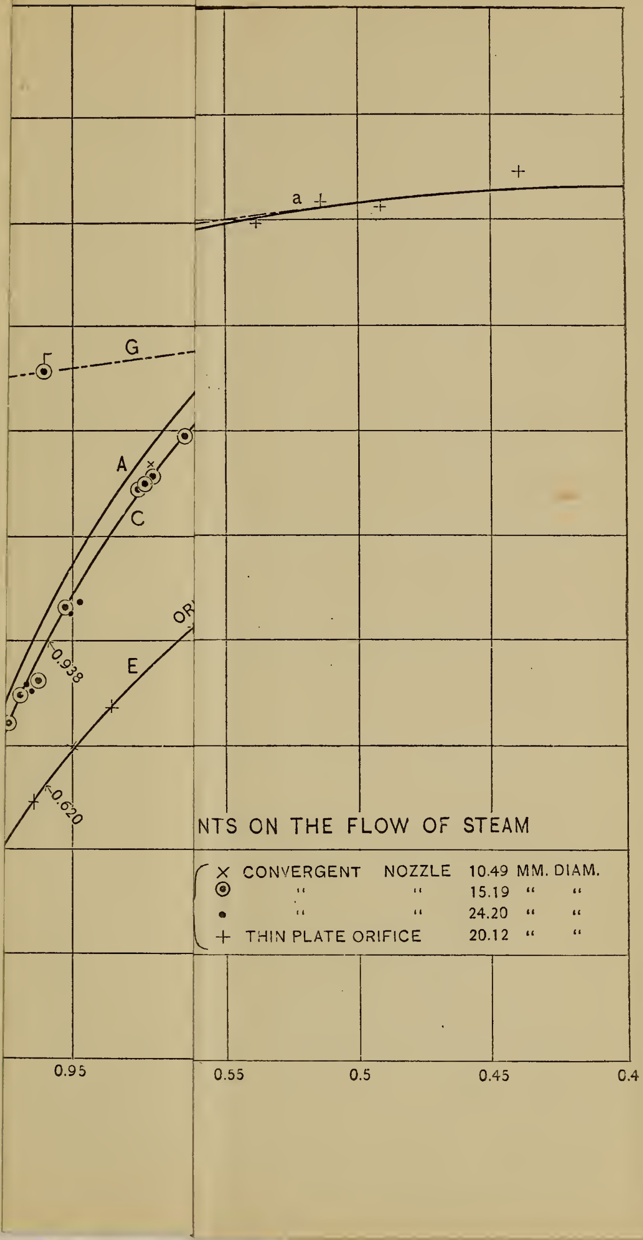
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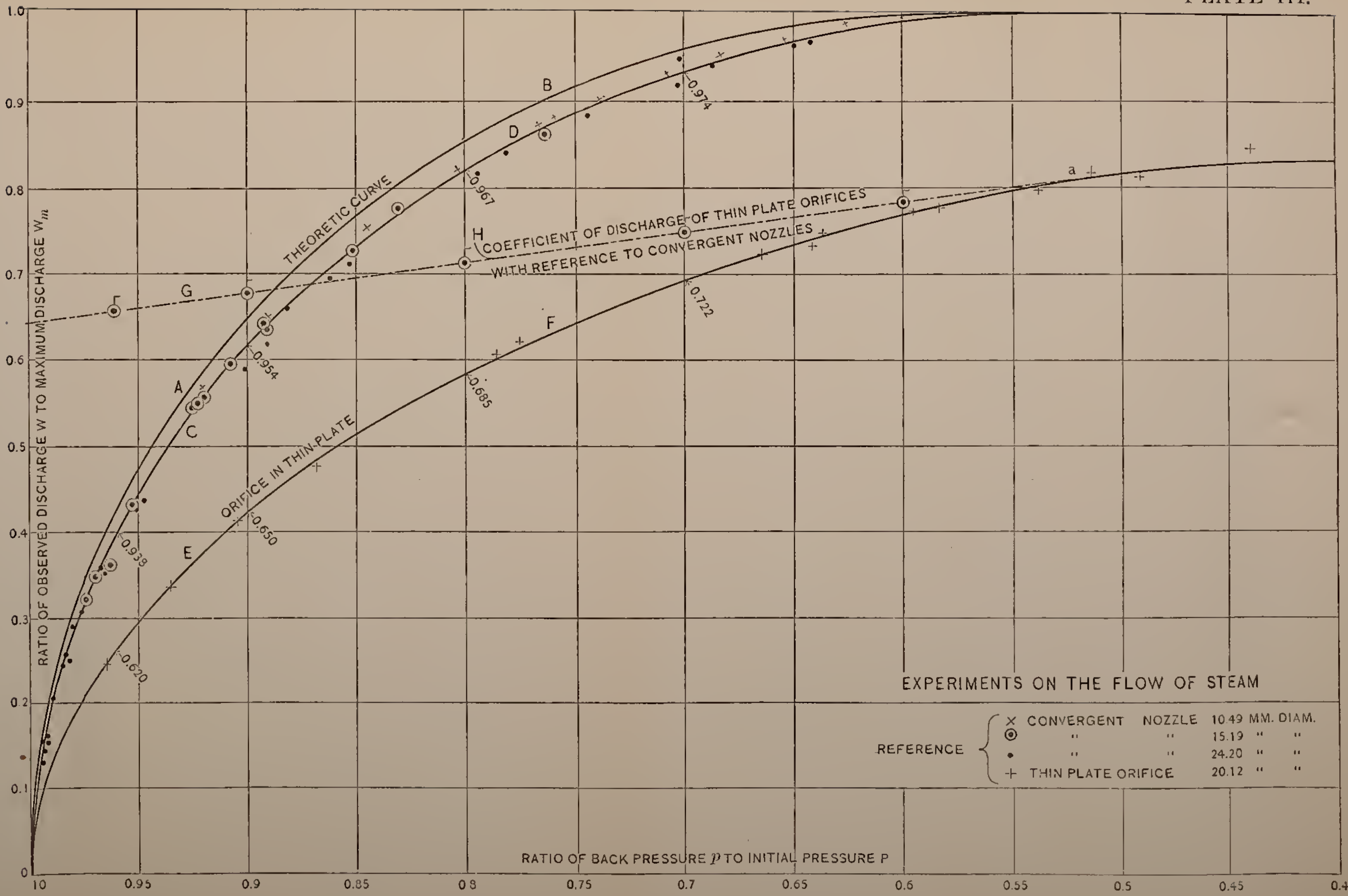
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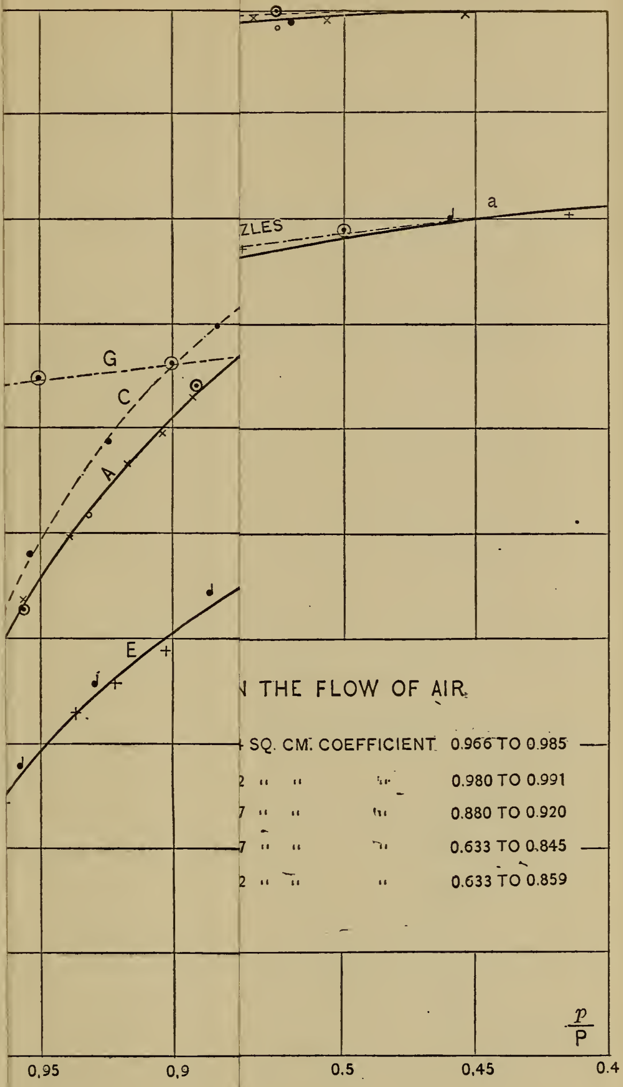
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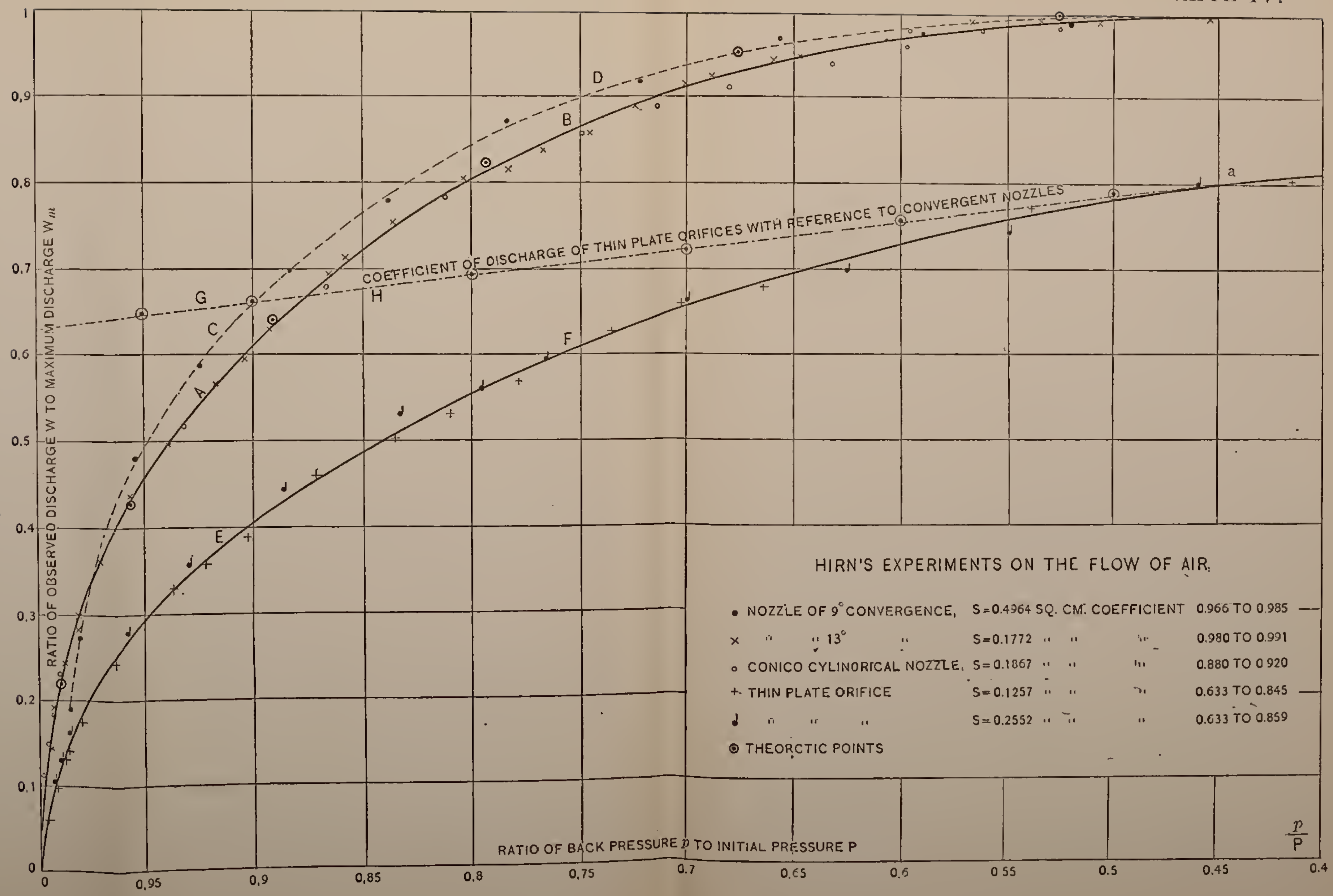
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